THE STEAM ENGINE

A CONCISE TREATISE FOR STUDENTS AND ENGINEERS

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PREFACE

This book is designed primarily for a text book, rather than as a work of reference for engineers. It has been the purpose of its author to explain the elementary principles of heat engines so that they might be readily understood by the average student and to show the application of these principles in the design, construction and operation of modern reciprocating engines.

Perhaps more attention has been paid to practical problems and less to theoretical ones, more attention to the questions of the engine user and less to those of the scientist. As far as possible, questions of economy of operation are settled by reference to the results of recent experiments under working conditions rather than by theoretical considerations. Some of these results are new and published here for the first time.

Attention is called to the chapter on Specifications and Costs as containing some new material.

It is hoped that the book does not lack anything which is really essential and that it contains nothing which can well be omitted.

C. H. B.

CONTENTS

INTRODUCTION.

Definitions.

Mass is quantity of matter and is usually measured by the attraction of gravity or weight at the surface of the earth. The algebraic expression for mass is

$$m = \frac{W}{g},$$

when $W =$ weight in pounds
and $g =$ acceleration due to gravity $= 32.2$ ft. per sec.

Since both W and g vary according to the distance from the surface of the earth, $\frac{W}{g}$ is a constant for any given mass, no matter what its location.

Force is that which tends to move or change the motion of mass and is usually measured by pounds, since the force of gravity is the best known and most universal of all forces.

A force may or may not change the motion of the mass on which it acts. In the former case the force is said to do *work* and the work is measured by the product of the force by the distance through which the mass is moved in a given time.

Force without motion, like "faith without works," is dead.

Work may be expressed as

	force $\times$ distance $= fs$
or as	unit pressure $\times$ volume $= pv.$
Since	unit pressure $\times$ area $=$ force
and	volume $\div$ area $=$ distance,
therefore	$fs = pa \times \frac{v}{a} = pv.$

In the case of circular motion,

	work $=$ moment $\times$ angular motion.
Since	moment $\div$ radius $=$ force
and	angular motion $\times$ radius $=$ distance,
therefore	$fs = \frac{T}{r} \times 2\pi Nr = 2\pi NT.$

Stress is an internal force exerted between the particles of a body and caused usually by the application of external forces.

Energy is the capacity for doing work, due either to the position or motion of a body, with reference to other bodies.

Static energy is energy due to position, as that in a hanging weight or a coiled spring.

Kinetic energy is that due to motion, as in falling water or a revolving fly wheel. Energy, like work, is measured in foot pounds.

Power is the rate of work or the amount of work done in a unit of time. It is usually expressed in foot pounds per minute or per second.

Horse Power is a convenient but rather arbitrary unit of power and represents 33,000 ft. lb. per min.

or 550 ft. lb. per sec.

It represents approximately the rate of work of a large horse in the ordinary treadmill.

Indicated Horse Power is the power of an engine, as shown by the indicator at the cylinder of an engine, and represents the gross work done by the steam or the energy delivered to the engine.

The i.h.p. of an engine is found from the formula

$$\text{h.p.} = \frac{PLAN}{33,000}$$

where

$P =$ mean pressure on square inch of piston.
$L =$ length of stroke in feet.
$A =$ area of piston in square inches.
$N =$ number of strokes per minute.

It is evident that the numerator gives the work done in ft. lb. per min.

Brake Horse Power is the power developed at the belt wheel of the engine and represents the net work done or the energy delivered by the engine. The efficiency of the engine as a machine is the ratio of the brake to the indicated horse power.

Since engines have been so much used for electrical work the electrical units have frequently been used. A *watt* is the unit of

electrical power, being the product of a volt by an ampere, and is equal to 0.737 ft. lb. per sec.
or 44.24 ft. lb. per min.
or $\frac{1}{746}$ of 1 h.p.

A *kilowatt* is one thousand watts and is equal to

$$\frac{1000}{746} = 1.34 \text{ h.p.}$$

Acceleration is the change of velocity of a body due to the action of an unbalanced force and is measured by the increase or decrease of velocity in a unit of time. Since the acceleration due to the force of gravity W is g ft. per sec., that produced by any force f will be proportional to $\frac{f}{W}$ or in symbols

$$a = \frac{fg}{W}$$

Conversely the force f required to produce an acceleration a in a mass weighing W is

$$f = \frac{W}{g} a$$

i.e. accelerating force = mass × acceleration.

Centrifugal Force is a measure of the resistance which any body offers to being deflected from a straight path. It is proved in works on mechanics that the rate of deflection or the radial acceleration is equal to $\frac{v^2}{r}$, where

v = velocity in ft. per sec.
and r = radius in ft.

The deviating force is accordingly

$$\text{mass} \times \text{acceleration} = \frac{W}{g}\,\frac{v^2}{r}$$

and the centrifugal force is equal and opposite to this.

Abbreviations.

Inches, in.; feet, ft.; yards, yd.; pounds, lb.; gallons, gal.; seconds, sec.; minutes, min.; hours, hr.; linear, lin.; square, sq.; cubic, cu.; Fahrenheit, Fahr.; Centigrade, Cent.; percentage, per cent; brake horse power, b.h.p.; electric horse power, e.h.p.; indicated horse power, i.h.p.; kilowatts, kw.; British thermal units, B.t.u.; diameter, diam.; horse power, h.p.

Notation—Area of section $= A$ sq. in.; breadth of section $= b$ in.; coefficient of friction $= f$; depth of section $= h$ in.; diameter of circular section $= d$ in.; entropy of liquid $= \theta$; entropy of vapor $= \phi$; heaviness, weight per cu. ft. $= d$; length of any member $= l$ in.; load or dead weight $= W$ lb.; mechanical equivalent of B.t.u. $= J$ ft. lb.; pressure of a gas $= p$ lb. per sq. ft.; ratios of expansion $= R$, r; specific heats of gases $= C$, c; temperature Fahrenheit $= t$, absolute $= T$; velocity $= v$ ft. per sec.; volume of a gas $= v$ cu. ft.

THE STEAM ENGINE

CHAPTER I.

ELEMENTARY PRINCIPLES

1. **Heat.**—Heat is believed to be one kind of motion of the molecules of matter and as such is one form of energy, since it combines the two factors of mass and motion.

It is the object of heat engines of all types to convert this molecular energy known as heat into the so-called mechanical work,—the motion of mass as a whole against external resistances. In this process matter in the gaseous form, such as air or steam, is best adapted to serve as a heat carrier, since it is thus most readily conveyed, compressed and expanded.

The air or steam receives from some convenient source heat which raises its temperature and pressure, and is conveyed in this condition by means of pipes to the working cylinder of the heat engine, where it is allowed to expand and push ahead of it some sort of a piston. This piston by means of suitable mechanism does work in the pumping of water, the lifting of weights or the turning of wheels. The air or steam after expanding is allowed to escape and the remainder of its heat energy goes to waste or is utilized in some other way than the doing of mechanical work.

In order to understand the conversion of heat into work it is necessary to know about the behavior of gases and vapors when heated. A so-called perfect or permanent gas is a substance which under ordinary circumstances is in the gaseous form, as common air or coal gas.

A vapor is a substance which, while in the gaseous condition, is near the border line between gases and liquids and is readily condensed by a slight change of conditions, as is the case with steam or ammonia gas.

2. **Temperature and Entropy.**—As heat is a form of energy

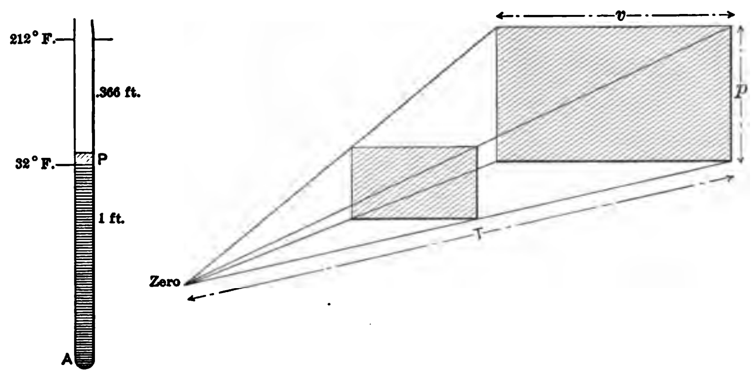
its use involves the presence of two factors, whose product is a measure of the transfer of energy. For example, in a hoisting machine we lift a weight through a distance; in a pump we cause a current or flow of liquid by means of a difference of pressure; in a dynamo we cause a current of electricity by a difference of a potential.

The product of the weight by the distance or of the pressure by the current is a measure of the work done.

The two factors in heat transfer are called temperature and entropy. Temperature is a measure of the intensity of the heat in a substance, as potential is of the electric intensity. A difference of temperature between different parts of a body tends to cause a flow of heat, as a difference of potential tends to cause an electric current. Entropy is the factor which measures the current or flow of heat as amperes measure the electric current.

Temperature can be measured directly while entropy cannot. (For a further discussion of the latter see Chapter III.)

Temperature is usually measured by the expansion or contraction of various substances under the influence of heat.



Figs. 1 and 2. Illustrating Absolute Zero.

3. **Air Thermometer.**—Let a tube of non-expansible glass, Fig. 1, contain 1 ft. of dry air at the temperature of melting ice, or 32 deg. Fahr., enclosed by a frictionless air-tight piston *P*. Then it can be shown by experiment that if the tem-

perature be raised to that of boiling water under atmospheric pressure, or 212 deg. Fahr., the volume of the air will increase until it fills 1.366 ft. of the tube; i.e. it expands 36.6 per cent.

It can be further shown that for each degree of heating or cooling the air expands or contracts a proportional amount.

To increase or diminish the volume 100 per cent will ac-accordingly require 180/0.366, or 492 deg. At a temperature of 492 deg. below freezing point of water, or 460 deg. below zero, the air would entirely disappear if this law continued to hold. As a matter of fact the air would liquefy before reaching this temperature and the law would change.

4. Absolute Temperature.—The point A on the air thermometer is called the absolute zero and is 460 deg. below the Fahrenheit zero. The exact location of the absolute zero for a perfect gas has been computed to be -460.66 deg. Fahr. The absolute temperature of a substance is that measured from the absolute zero and is found by adding 460.66 deg. to its Fahrenheit temperature.

Hereafter the symbol t will be used for temperature Fahrenheit, and T for absolute temperature.

5. Change of Pressure and Volume.—Both the pressure and volume of a gas may change with the temperature. Let p, v and T represent respectively the pressure, volume and absolute temperature of a perfect gas; then can it be shown by experiment that within observed limits pv is proportional to T (see Fig. 2)

or $$pv = kT, \text{ or } \frac{pv}{T} = \frac{p_1 v_1}{T_1} \text{ a constant.}$$

Let $p_1 v_1$ be the product of the pressure and volume of a gas at 212 deg. Fahr., and $p_0 v_0$ the same at 32 deg. Fahr., then it follows from what goes before, that

$$\frac{p_1 v_1}{p_0 v_0} = \frac{T_1}{T_0} = 1.366$$

6. Quantities of Heat.—Quantities of heat are measured by means of the various changes in bodies which accompany the

transfer of heat, as change of temperature, change of volume, melting and evaporation.

Equal changes of temperature do not necessarily imply equal quantities of heat transferred; for instance the quantity of heat required to raise a pound of water 1 deg. will raise a pound of iron about 9 deg.

7. Unit of Heat.—The British thermal unit is the quantity of heat necessary to raise 1 lb. of distilled water 1 deg. Fahr. from 39 deg. Fahr., its temperature of greatest density.

This unit will be denoted by the symbol B.t.u. in these notes.

The specific heat of water at 39 deg. Fahr. is taken as unity; the specific heat of any other substance may therefore be expressed as the fraction of a thermal unit required to raise one pound of that substance 1 deg. Fahr. The specific heats of nearly all substances are less than unity.

8. Latent Heat.—Latent heat is heat which produces some other change in the substance than that of temperature. There are latent heats of expansion, of fusion and of evaporation. In each of these the heat which is given to the body produces some molecular change.

For instance, to raise the temperature of a pound of air in a closed vessel 1 deg. Fahr., will require 0.169 B.t.u.; but to raise the temperature the same amount, when the air is allowed to expand at a constant pressure, will require 0.238 B.t.u. The difference, 0.069 B.t.u., is the latent heat of expansion of air.

If heat be continually applied to a pound of ice, the results will be as follows: From original temperature to 32 deg. Fahr. the specific heat will be 0.504. The latent heat of fusion, at 32 deg. will be 142 B.t.u. nearly, the temperature remaining the same until the ice is all melted.

From 32 deg. to 212 deg. Fahr. the specific heat will be nearly unity, being slightly greater near 212 deg. The latent heat of evaporation at 212 deg. will be 966 B.t.u., the temperature remaining constant until the water is all evaporated.

Above 212 deg. the specific heat will vary from 0.5 to 0.7.

Fig. 3 illustrates the heating of water and of dry air.

9. Thermodynamics.—The science which treats of the relations between heat and mechanical energy is called "thermo-

dynamics"; i. e. "the science of heat power." Joule, an English experimenter, found, by stirring water, that 772 ft. lb. of mechanical energy were used in raising the temperature of a pound of water 1 deg. Fahr. and this number 772 was long known as "Joule's Equivalent." Subsequent experiments by the late Pro-

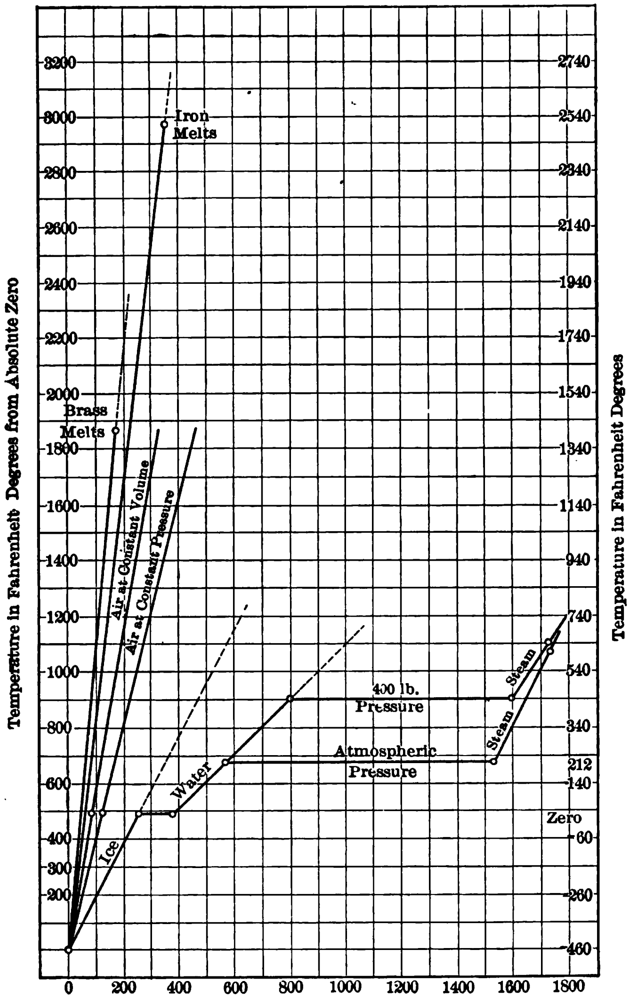


Fig. 3. Changes when Heating Dry Air and Water.

fessor Rowland have shown 778 to be nearer the correct value. We will accordingly say that our B.t.u. is equivalent to 778 ft. lb. and will denote this ratio by the letter J. As has been already noticed, heat is usually converted into work by means of the expansion of a heated gas which loses its heat as it does work in expanding.

10. **Expansion of Gases.**—According to the law explained in Art. 5 the pressure of a gas usually diminishes as the volume increases, the temperature either remaining constant or gradually decreasing. The changes of pressure and volume can be shown graphically as in Fig. 4, volumes being measured on OX

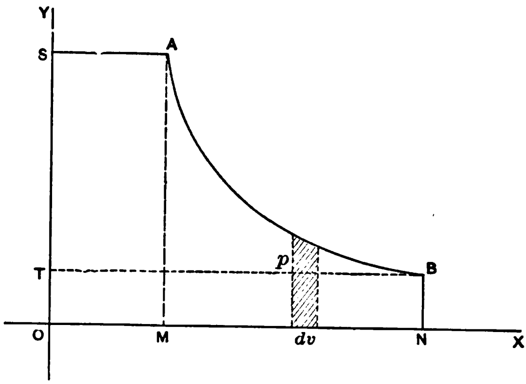


Fig. 4. Pressure-Volume Diagram.

and pressures on OY. If A is the original-state point of the gas, its volume in cubic feet is measured by OM and its pressure in pounds per square foot by OS. If the volume increase to ON, its pressure will fall to OT, the relations between p and v depending on the temperatures.

Since the points on the line AB show graphically the changes of pressure and volume, the area $ABNMA$ under the curve represents the work done by the gas in expanding. For, let p be the pressure at any instant and dv the increase of volume, in time dt, then will pdv equal the increase of work and $\int pdv$ between A and B equal the total work.

11. Isothermal Expansion.—When a gas expands at a constant temperature it is said to expand isothermally and the equation given in Art. 5 resolves itself into $pv = c$, where c is a constant depending upon the characteristics of the gas.

In order to have a constant temperature in an expanding gas it is necessary to supply heat to it from outside to compensate for the heat converted into work. This may be done in a steam engine by means of a steam jacket around the working cylinder.

The curve of pressures and volumes for isothermal expansion is an equilateral hyperbola referred to its asymptotes as axes, since the equation of such a curve is $xy =$ a constant. The curve may be constructed by means of rectangles of equal area, or more readily by the method shown in Fig. 5.

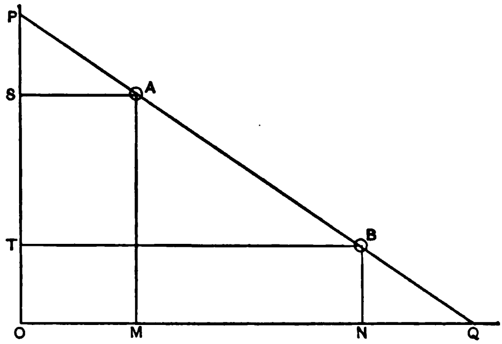


Fig. 5. Construction of Equilateral Hyperbola.

Let A be a given point in the curve. Draw any oblique line PAQ cutting the two axes in P and Q. It is a property of the curve that the two intercepts of a secant included between the curve and the axes are equal. Therefore lay off $QB = PA$ and B will be another point on the curve. Other points may be found in a similar manner.

The area under an isothermal curve represents graphically the work done by expansion of a gas at constant temperature.

Let this area $ABNM$, Fig. 4, be called A, and let $p =$ the pressure at any instant in pounds per square foot, $v =$ volume in

cubic feet, and dv = infinitesimal change of volume. Then by calculus:

$$A = \int_{v_1}^{v_2} p\,dv$$

But $$pv = p_1 v_1 \text{ and } p = \frac{p_1 v_1}{v}$$

$$A = p_1 v_1 \int_{v_1}^{v_2} \frac{dv}{v} = p_1 v_1 (\log_e v_2 - \log_e v_1)$$

$$= p_1 v_1 \log_e \frac{v_2}{v_1} = kT_1 \log_e \frac{v_2}{v_1} \text{ for 1 lb. of gas.} \qquad \text{(a)}$$

The whole area of the diagram $SABNO$

$$= p_1 v_1 + A = p_1 v_1 \left(1 + \log_e \frac{v_2}{v_1}\right) \qquad (1)$$

This whole area represents the work done in foot pounds by admitting the volume of gas SA to a cylinder and then expanding it.

The mean pressure of the gas =

$$p_m = \frac{\text{area}}{ON} = \frac{p_1 v_1}{v_2}\left(1 + \log_e \frac{v_2}{v_1}\right) = p_2\left(1 + \log_e \frac{v_2}{v_1}\right) \qquad (2)$$

12. Adiabatic Expansion.—A gas expanding in a non-conducting cylinder is said to expand adiabatically; i. e. "no heat passing through" and its temperature will fall during the expansion since no heat is supplied to take the place of that converted into work.

This condition is probably never realized in practice, but is approximated when a gas expands suddenly, as there is then little time for the conduction of heat. Adiabatic expansion is a convenient assumption as a starting point in studying the work of expansion of gases. This will be more fully explained in Chapter III. The curve of pressures and volumes is an hyperbola and similar to that in Fig. 5, but falling more rapidly with increase of volume on account of the falling temperature. Its

equation can be proved to be $pv^n =$ a constant. The value of n for different gases is calculated in Art. 22. The area under the curve representing the work of expansion is calculated as follows:

In this case

$$pv^n = p_1v_1^n$$

and $$p = p_1v_1^n v^{-n}$$

Substituting this value of p in the equation

$$A = \int_{v_1}^{v_2} pdv$$

and we have $$A = p_1v_1^n \int_{v_1}^{v_2} v^{-n}\, dv$$

$$A = p_1v_1^n \frac{v_2^{1-n} - v_1^{1-n}}{1-n}$$

$$A = p_1v_1^n \frac{v_1^{1-n} - v_2^{1-n}}{n-1} \tag{3}$$

This expression may be reduced to either of three forms

$$A = \frac{p_1v_1 - p_2v_2}{n-1} = \frac{k}{n-1}(T_1 - T_2) \text{ for 1 lb. of gas}$$

$$\text{or} \quad \frac{p_1v_1}{n-1}\left\{1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right\} \tag{4}$$

Adding the expression p_1v_1 will give the whole area of the diagram $=\frac{np_1v_1 - p_2v_2}{n-1}$ The mean pressure equals this last divided by $v_2 = \frac{np_1v_1 - p_2v_2}{(n-1)v_2}$

In terms of the pressures equation (4) reads:

$$A = \frac{p_1v_1}{n-1}\left\{1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right\} \tag{4a}$$

Let gas expand in a non-conducting cylinder from a pres-

sure, volume and temperature denoted by p_1, v_1 and t_1 to those denoted by p_2, v_2 and t_2.

Then
$$p_2 v_2^{n} = p_1 v_1^{n}$$

$$p_2 v_2 = p_1 v_1 \left(\frac{v_1}{v_2}\right)^{n-1}$$

But
$$pv = kT$$

$$kT_2 = kT_1 \left(\frac{v_1}{v_2}\right)^{n-1}$$

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{n-1} = T_1\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} \tag{5}$$

or
$$\frac{v_1}{v_2} = \left(\frac{T_2}{T_1}\right)^{\frac{1}{n-1}} \tag{6}$$

The formulas derived in this and the preceding article are applicable to the so-called permanent gases such as are used in air and gas engines.

13. Steam.—Steam is the vapor of water and may be either saturated or superheated.

Saturated steam is vapor in contact with the water from which it was formed and is ready to condense at the slightest change of condition. Water boils at some definite temperature corresponding to the pressure and this temperature remains the same until the water has all turned to steam. If the pressure increases the temperature of boiling increases. A sudden decrease of pressure will cause rapid boiling until the added vapor restores the normal pressure; a sudden increase of pressure will cause condensation of some of the vapor until again equilibrium is established. Water and steam under these conditions are in unstable equilibrium and ready to change their state at the slightest disturbance of the prevailing conditions.

Superheated steam is that which has been removed from contact with the water and heated to a temperature greater than that corresponding to its pressure. See Chapter X. Such steam is in the condition of a permanent gas and will not readily condense. The steam entering an engine is sometimes superheated, but

usually becomes saturated at or before the beginning of expansion.

The heating and evaporating of water has been briefly explained in Art. 8 and will be further described in Chapter IV.

Tables giving the pressures and temperatures of saturated steam and the quantities of heat required are given at the end of the book.

The curve of pressures and volumes for saturated steam is approximately $pv^{1.06} =$ a constant, falling slightly below the isothermal curve $pv = c$ as expansion continues.

14. Indicator Diagrams.—The changes of pressure and volume of the gas or vapor in an engine cylinder are recorded graphically by the indicator, an instrument which is described in detail in Chapter VI. It is sufficient for our present purpose to say that this instrument draws upon paper a diagram which

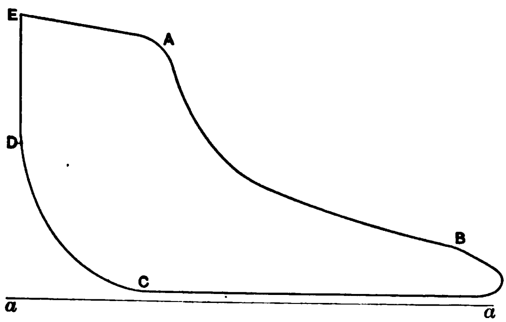


Fig. 6. Typical Indicator Diagram.

shows the pressure and volume of the working fluid inside the cylinder throughout a complete revolution of the engine. A typical indicator diagram for a steam engine is shown in Fig. 6. *EA* shows the admission of steam to the cylinder at an approximately constant pressure. *A* is the point where the admission of the steam is checked by the closing of the inlet valve. This is commonly known as the point of cut-off. *AB* is the expansion of the steam after cut-off and is more or less like an hyperbola. *B* is the point where the outlet or exhaust valve opens allowing the steam to flow away from the cylinder into the exhaust pipe, and is called the point of release.

The curve from B to C shows the escape of the steam from the cylinder as the engine piston finishes its stroke and returns. C is the point where the outlet valve closes and prevents further escape of steam, known as the compression point. CD is the curve caused by the compression of the residual steam into the small clearance space at the end of the cylinder. From D to E is the rise of pressure caused by the admission of fresh steam. Line *aa* indicates the pressure of the atmosphere.

The area underneath the admission and expansion lines represents the work done by the steam. The area under the exhaust and compression lines represents work done by the piston on the steam and may be regarded as negative. The difference of these two areas or the area of the diagram $EABCD$, represents the net work done by the steam. This area divided by the length of the diagram gives the mean height, or, as it is usually called, the *mean effective pressure,* so much used in calculating the power of the engine.

PROBLEMS.

1. If a pound of dry air at 32 deg. Fahr. occupies 12.387 cu. ft. under atmospheric pressure of 14.7 lb. per sq. in., what will be its volume at 46 lb. per sq. in. and 64 deg. Fahr.?

2. Find the number of B.t.u. required to convert a pound of ice at 0 deg. Fahr. into steam having 50 deg. superheat at atmospheric pressure.

3. Find the number of B.t.u. required to heat a pound of dry air through the same range of temperature as in Problem 2: (a) at constant pressure; (b) at constant volume.

4. A pound of dry air at 60 deg. Fahr. and a pressure of 75 lb. absolute expands to three times its initial volume at a constant temperature. Find: (a) the initial and final volumes and the final pressure; (b) the work of expansion in foot pounds; (c) the total work and the mean pressure.

5. A pound of dry air under the same initial conditions as in Problem 4 expands adiabatically to three times the initial volume. Find: (a) the final pressure and temperature, assuming $n = 1.4$;

(b) the work of expansion; (c) the total work and the mean pressure.

6. The indicator card of an engine is 3.2 in. long and its area is 2.4 sq. in. The spring used is such that an inch of height represents 60 lb. per sq. in. Find the mean height in inches and the mean effective pressure in pounds. Find the i.h.p. of the engine if the diameter of the cylinder is 16 in., the length of the stroke 2 ft. and the number of revolutions 120 per minute.

CHAPTER II.

THE SIMPLE STEAM ENGINE.

15. Description.—The steam engine reduced to its simplest form consists of the following parts, each having its definite function. (See Fig. 7.)

1.	Steam pipe.	16.	Out-board bearing.
2.	Throttle valve.	17.	Fly wheel.
3.	Steam chest.	18.	Eccentric.
4.	Admission valves.	19.	Eccentric rod.
5.	Exhaust valves.	20.	Valve stem.
6.	Cylinder.	21.	Rock-arm.
7.	Piston.	22.	Governor.
8.	Piston rod.	23.	Exhaust pipe.
9.	Stuffing box.	24.	Condenser.
10.	Crosshead.	25.	Air pump.
11.	Guides.	26.	Hot-well.
12.	Connecting rod.	27.	Injection pipe.
13.	Crank.	28.	Discharge pipe.
14.	Shaft.	29.	Steam pump.
15.	Main pedestal.	30.	Engine frame.

The steam pipe (1) carrying steam at high pressure from the boiler to the engine. The throttle valve (2) regulating the admission of steam to the steam chest of the engine. The steam chest (3) forming a sort of ante-chamber to the cylinder where the steam is held until wanted. The admission valves (4) by which the steam is admitted through ports to the cylinder. The exhaust valves (5) by which steam is released from the cylinder after it has been used. Valves (4) and (5) are sometimes included in one valve as in so-called "slide-valve" engines. The cylinder (6) in which the steam expands and does its work. The piston (7) which is pushed back and forth by the steam entering the two ends of the cylinder alternately.

The piston rod (8) which transmits the motion of the piston to the crosshead. The stuffing-box and gland (9) which prevent the escape of steam around the rod. The crosshead and wrist pin

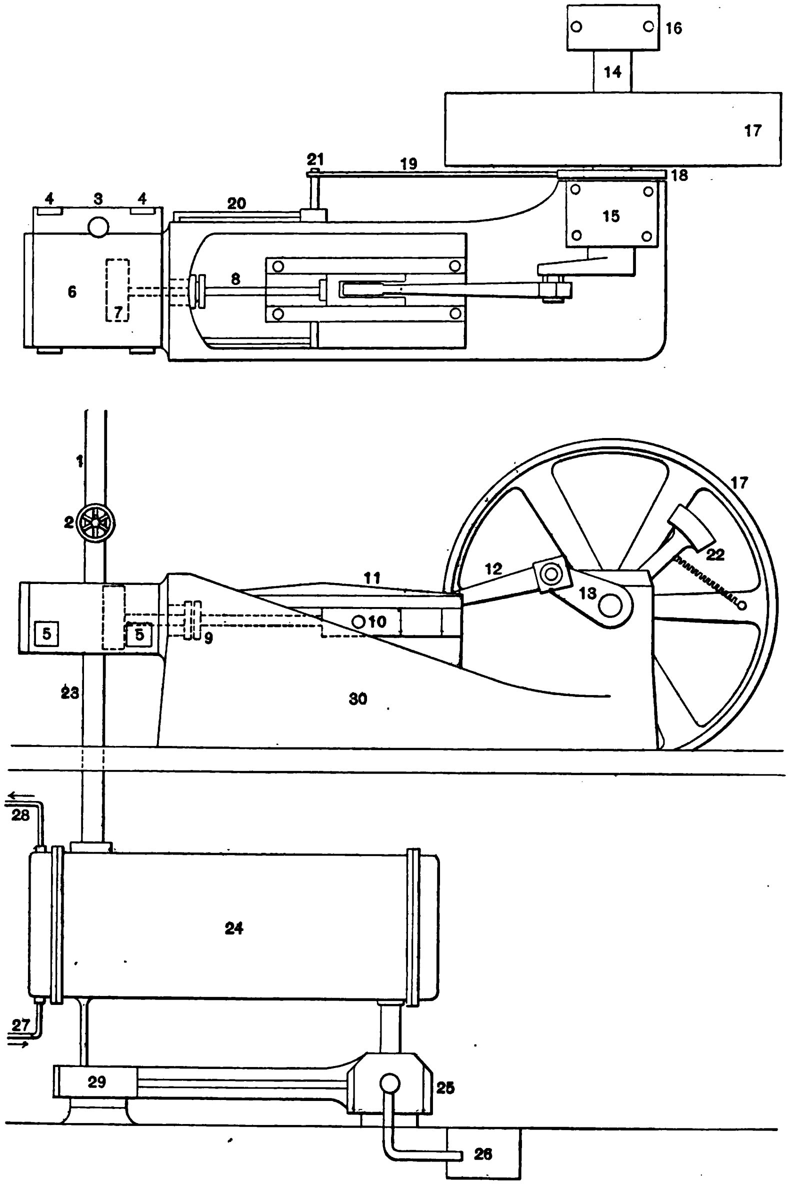


Fig. 7. Simple Steam Engine.

(10) serving as a connection between the piston rod and connecting rod. The guides (11) which insure a straight line motion for the crosshead. The connecting rod (12) which converts this straight line motion into the rotary motion of the crank. (The ends of the connecting rod are usually called stub-ends.) The crank (13) and its pin, serving to turn the shaft in a continuous rotary motion when impelled by the connecting rod. The shaft (14) which carries all the rotary parts. The main pedestal or pillow-block (15) supporting that end of the shaft nearest the crank. The out-board bearing (16) at the other end of the shaft.

The fly wheel (17) rotating with the shaft and serving to equalize the irregular turning forces of the crank. This may also be used as a belt wheel for transmitting the power from the engine to shafting.

The eccentric (18) which imparts motion to the valves through the eccentric rod (19) and the valve stem (20). There will be one or more eccentrics according to the type of valve motion and the connection with the valves may be through a rock-arm (21) or a crosshead and guides.

The governor (22) regulates the speed of the engine. It may be a revolving pendulum or it may be a shaft-governor as shown in the figure, consisting of revolving weights located inside the fly wheel and controlled by springs.

In case the engine is non-condensing the exhaust pipe (23) leads directly to the open air or to heating coils. If the engine is of the condensing class the exhaust is led into a condenser (24). The condenser shown in the figure is of the surface type, where the steam is kept separate from the condensing water. Surface condensers are used with marine engines where the condensing water is salt. Stationary engines are usually equipped with jet condensers, in which the steam and water combine and are pumped out together. The surface condenser is filled with banks of brass tubes around which the steam passes and through which the water circulates.

In the illustration the condensed steam is drawn out by means of the air pump (25) which maintains a vacuum in the condenser and discharges the condensed steam into the hot-well (26). The cold water is forced by a pump or by the pressure in the

mains through the injection pipe (27) into the condenser and is ejected through the discharge pipe (28). The air pump is usually driven by a direct-acting steam pump (29).

A more detailed description of the various forms of condensing apparatus will be found in Chapter XI.

16. Forces and Stresses.—The steam engine furnishes one of the best examples of machine design, since it is usually possible to determine beforehand the nature and extent of the impressed forces. The engine also illustrates clearly the action of reciprocating and rotating parts and for that reason furnishes a good example in applied mechanics. A study of the several forces and stresses which are involved in the design and operation of the steam engine will be made in subsequent chapters, and only a brief review of the subject will be attempted here.

The cylinder walls are subjected to tensile and transverse stresses, due to cooling of the metal and to the internal pressure of the steam.

The piston rod suffers alternate tension and compression and is usually regarded as a column with one flat and one round end.

The connecting rod is subjected to similar forces, but must be regarded as round at both ends and is also affected somewhat by the swinging motion in high-speed engines.

The piston and its rod, the crosshead and the connecting rod are together called the *reciprocating parts* and tend by their inertia to shake the engine as they are alternately accelerated and retarded. This action is partially neutralized by the counterbalance on the crank, but cannot be entirely eliminated. The wrist pin, crank pin and the main journal of the shaft are all exposed to the full pressure of the piston and are liable to wear or heat unless made of liberal proportions.

The pressure caused by the weight of the fly wheel must also be considered in designing the shaft journals. The shaft is an example of a machine member exposed to both twisting and bending. The rim of the fly wheel is subjected to tension and to bending on account of its own centrifugal force and must be carefully designed.

The crank itself and the arms of the fly wheel are examples of cantilevers or beams exposed to bending.

17. In General.—The frame has to fulfill the twofold function of withstanding the stresses due to the steam pressure and of resisting by its mass the tendency to vibration caused by the rapid motion of the reciprocating parts. Three types of frame are in general use: (a) The *curved Tangye bed* similar to that shown in Fig. 7, depending on its mass for resistance to the applied forces and appropriate for engines which have a high rotative speed; (b) the *straight girder* frame so much used in the relatively slow moving Corliss engine; (c) the *A-frame,* which has two straight girders running from the cylinder to the two shaft bearings and which is more often seen in vertical engines (see Fig. 168).

Engines are known as *vertical* or *horizontal* according to the position of the axis of the cylinder.

A *side-crank* engine is one in which the crank is at one end of the shaft as in Fig. 7. A *center-crank* engine is one in which the crank is near the center of the shaft between the two journals (see Fig. 169).

A side-crank engine is denominated *right-hand* or *left-hand* according as the shaft extends to the right or left hand of the observer when he stands at the cylinder and faces the crank.

An engine is said to *throw over* when the connecting rod rises as the crosshead begins to move from the outer end of its stroke towards the crank; it is said to *throw under* when it moves in the opposite direction.

In Fig. 7 the engine would be throwing over if the fly wheel were turning in a right-hand direction. This is the direction in which most engines are designed to turn. An overthrow engine has a downward pressure on the crosshead guides, insuring greater rigidity and better lubrication than when the pressure is on the upper guide.

Engines are sometimes classified as high-speed, medium-speed and low-speed. This classification refers entirely to the speed of rotation since the average speed of the piston is nearly the same in all classes, varying perhaps from 500 to 700 ft. per minute.

A *high-speed* engine is characterized by a short stroke and high rotative speed, and is usually equipped with a shaft governor and a positive valve movement. The *slow-speed* engines on

the other hand have relatively long strokes and slow rotative speeds and are usually fitted with pendulum governors and Corliss detachable valve motions (see Chapter VII.). No fixed lines can be drawn between these different classes, but in general it may be said that engines which make from 200 to 300 revolutions per minute are high-speed, while those which make less than 100 are low-speed.

CHAPTER III.

THE THERMODYNAMICS OF AIR.

18. General Equations.—In this chapter air will be considered as the working medium, but it may be understood that the equations used will serve as well for any of the so-called permanent gases.

Four factors are always present and subject to variation in problems of the thermodynamics of a gas: pressure, volume, temperature and entropy.

The product of the first two will be a measure of the mechanical energy available and that of the last two a measure of the heat energy. Any or all of these may vary according to the conditions imposed.

The pressure of the air is measured by comparison with springs or weights; the volume by measurement of the space occupied; the temperature by the expansion of known substances, such as mercury or copper; the entropy indirectly by measuring the heat transferred and the temperature. Since heat energy is the product of the temperature and entropy of a gas, the entropy may be determined by dividing the heat expressed in heat units by the absolute temperature in degrees Fahrenheit. We will denote the factors by the following letters:

p=pressure in lb. per sq. ft.
v=volume in cu. ft.
T=absolute temperature in deg. Fahr.
ϕ=entropy in units.
H=heat energy in heat units.
JH= " " " ft. lb.
W=mechanical energy in ft. lb.
Then will

$$W=pv \text{ and } H=T\phi$$

The specific heat of air at a constant volume is 0.169; that is, it takes 0.169 B.t.u. to heat a pound of air 1 deg. Fahr. If the air is allowed to expand at a constant pressure, heat is used in the work of expansion and 0.238 B.t.u. will be required. If ex-

pressed in foot pounds these quantities will be 131.5 ft. lb. and 184.8 ft. lb. respectively.

The latter quantities will hereafter be designated by the symbols C_v for constant volume and C_p for constant pressure.

Let us first consider the general case of air which is receiving heat from some outside source and at the same time doing mechanical work, and in which pressure, volume, temperature and entropy are all changing. We can then get a general equation from which special equations can be derived to suit various conditions.

When air is receiving heat in this way a part of the heat is used in raising the temperature, doing internal work by increasing the molecular velocity. If the gas is expanding, the remainder of the heat is converted into the mechanical energy of expansion or external work.

Let a minute interval of time be considered so that the heat received is dH heat units and the changes of condition can be denoted by dp, dv, dT and $d\phi$. If the quantity of air be one pound the internal work done will be $C_v\,dT$ expressed in foot pounds and the external work will be pdv. (See Art. 11.) The fundamental equation is then:

$$JdH = JTd\phi = C_v dt + pdv \tag{a}$$

($dH = Td\phi$, from the definition of entropy, just as $dW = pdv$. See the shaded area in Fig. 8.)

Dividing both members of the equation by T,

$$Jd\phi = C_v\frac{dt}{T} + \frac{pdv}{T} \tag{b}$$

By Art. 5, pv is proportional to T or $pv = kT$. A pound of dry air at atmospheric pressure, or 14.7 lb. sq. in., and at the temperature 32 deg. Fahr., has a volume of 12.387 cu. ft.

Therefore, $$k = \frac{14.7 \times 144 \times 12.387}{492} = 53.27$$

Substituting $\frac{k}{v}$ for $\frac{p}{T}$ in (b),

$$Jd\phi = C_v\frac{dT}{T} + \frac{k\,dv}{v} \tag{c}$$

a form which can be integrated.

Integrating between corresponding limits,

$$J(\phi_2-\phi_1)=C_v \; log_e \frac{T_2}{T_1}+k \; log_e \frac{v_2}{v_1} \qquad (7)$$

This equation gives the change in entropy for any change of temperature and volume.

The expression $\frac{v_2}{v_1}$ is called the ratio of expansion and is denoted by R.

Equation (7) can now be modified to suit various sets of conditions.

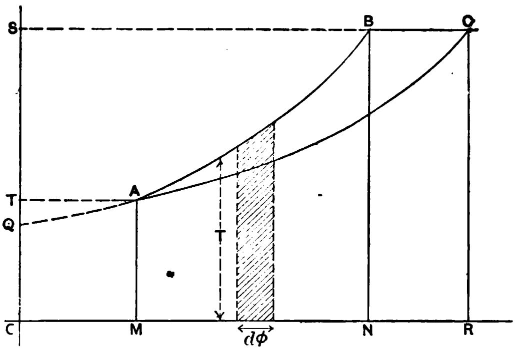


Fig. 8. Heat Changes at Constant Volume and at Constant Pressure.

19. Heating at Constant Volume.—In this case, since v is constant, no external work is done, and $pdv = o$, whence (7) reduces to

$$J(\phi_2-\phi_1)=C_v \log_e \frac{T_2}{T_1}=C_v(\log_e T_2-\log_e T_1) \qquad (8)$$

In representing heat equations graphically, T is used for ordinates and ϕ for abscissas. Since only the increase of ϕ is of importance, the location of the zero is immaterial, but it is usually taken at the value of ϕ for 32 deg. Fahr. The zero for T must be the absolute zero of the Fahrenheit scale.

In Fig. 8 let $OR=T_1$ and $OS=T_2$ and $OM=\phi_1$.

By plotting various values of T and ϕ from equation (8) we

may get points on the logarithmic curve AB and construct the curve. Then may $BN=T_2$ and $MN=(\phi_2-\phi_1)$. The area under the curve or $ABNM$ represents $\int Td\phi$ or the heat received by the air. This is equal to $C_v\ (T_2-T_1)$ ft. lb. where $C_v=131.5$ (Art. 18). The heat in the air at A above 32 deg. Fahr. is measured by the area $AMOQ$, where Q is the intersection of the curve BA, produced, with the Y-axis.

20. Heating at Constant Pressure.—If air receives heat and its pressure remains constant, its volume must increase, since $pv = kT$. This means that some of the heat will be used in the work of expansion.

If p is constant

$$\frac{v_2}{v_1}=\frac{T_2}{T_1}$$

Substituting this value in (7):

$$J(\phi_2-\phi_1)=C_v \log_e \frac{T_2}{T_1}+k \log_e \frac{T_2}{T_1}$$

$$=C_p \log_e \frac{T_2}{T_1} \tag{9}$$

where $C_p=184.8$ (Art. 18).

This is the equation of a second logarithmic curve AC in Fig. 8, having larger values of ϕ than the first curve. MR is the new value of $\phi_2-\phi_1$ and the area under the curve or $ACRM$, is now equal to $184.8\ (T_2-T_1)$. As $C_p-C_v=k$, the difference between the two areas is $k\ (T_2-T_1)=53.3\ (T_2-T_1)$, or the additional heat used in expanding the air.

21. Isothermal Expansion.—If the air is so heated as to expand at a constant temperature and do mechanical work, the heat received must equal the external work done (see Art. 11). In the general equation the first term disappears and there remains

$$J\ (\phi_2-\phi_1)=k \log_e \frac{v_2}{v_1} \tag{10}$$

As the temperature remains constant at T_1, the heat diagram

is a rectangle $ABNM$ (Fig. 9) and the heat received $= T_1(\phi_2 - \phi_1)$. Multiplying both sides of (10) by T_1,

$$JT_1(\phi_2-\phi_1)=kT_1 \log_e \frac{v_2}{v_1} = p_1 v_1 \log_e \frac{v_2}{v_1} \qquad (11)$$

By Art. 11, this is equal to the mechanical work done in expanding. The condition of the air as to heat energy therefore remains the same during the expansion.

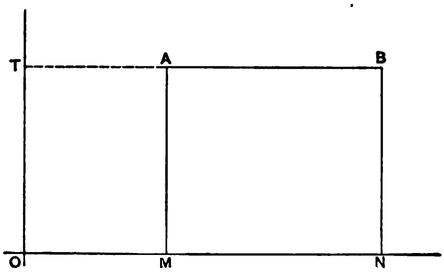


Fig. 9. Heat Diagram, Isothermal Expansion.

22. **Adiabatic Expansion.**—If the air expand doing work in a non-conducting cylinder, there will be no flow of heat, and the first member of equation (7) reduces to zero. Hence

$$0=C\,v \log_e \frac{T_2}{T_1} + k \log_e \frac{v_2}{v_1} \qquad (12)$$

If the air is assumed to expand, $\log_e \frac{v_2}{v_1}$ is positive and therefore $\log_e \frac{T_2}{T_1}$ must be negative, i. e. the temperature is falling. This corresponds with the assumption made in Art. 12.

In Fig. 9 adiabatic expansion would be represented by the vertical line AM, a fall of temperature without flow of heat. All the mechanical work is done at the expense of the internal energy.

Isothermal expansion may be compared to a business where the receipts equal the expenditures and the capital is not impaired. In adiabatic expansion there are no receipts and the expenditures reduce the capital.

The loss of internal energy due to fall of temperature equals $C_v(T_1-T_2)$ and from Art. 12 the work of expansion equals $\frac{p_1v_1-p_2v_2}{n-1}$. Equating these two expressions and substituting kT for pv,

$$C_v(T_1-T_2)=\frac{k}{n-1}(T_1-T_2)$$

$$n=\frac{C_v+k}{C_v}=\frac{C_p}{C_v} \tag{13}$$

This is the ratio of specific heat at constant pressure to that at constant volume, and may be denoted by r. Its value is as follows:

Dry air	1.405
Saturated air	1.2
Superheated steam	1.3
Saturated steam	1.11
Ammonia gas	1.27

The equation of the adiabatic curve may then be written

$$pv^r=p_1v_1^r \tag{14}$$

23. There are three ways in which energy may be transferred to or from the working cylinder of an engine which contains an expanding gas: (a) Through the inlet and outlet valves; (b) through the cylinder walls; (c) through the piston rod. The first two, (a) and (b), are transfers of heat energy, and (c) is a transfer of mechanical energy. The transfer (a) can occur only before cut-off and after release, while (b) and (c) may occur at any time. In isothermal expansion (b) supplies the heat needed for (c). In adiabatic expansion (c) must be taken from (a) as (b) is zero. In isodynamic expansion there is no piston rod and therefore no heat escapes in this way.*

24. Total Heat of Air.—Let air of pressure and volume p_1 and v_1 expand indefinitely in a non-conducting cylinder, doing

*By an isodynamic expansion is meant expansion of a gas from a high pressure cylinder into an adjacent low pressure cylinder without any expenditure of mechanical energy.

external work at the expense of its internal energy. Then will the indefinite area *ARXM* under the adiabatic curve *AR,* Fig. 10, represent the external work done and therefore the heat expended.

If the gas be supposed to expand to infinity, then will its temperature reach absolute zero, and

$$T_2 = T_1 \left(\frac{v_1}{\infty}\right)^{r-1} = 0$$

Consequently, all its internal heat will have been converted into work represented by the area under the expansion curve.

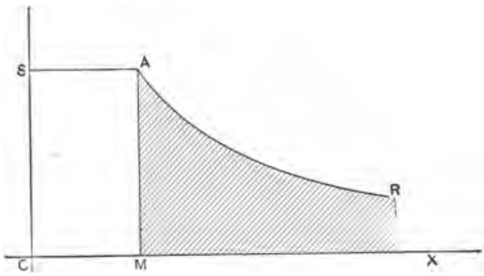


Fig. 10. Total Heat Shown by Work Diagram.

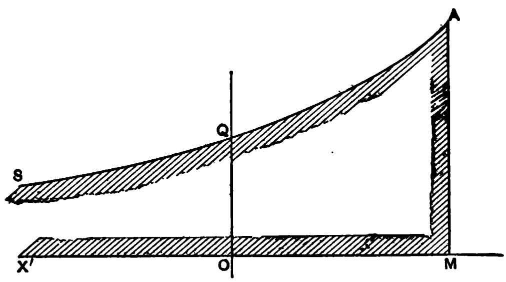


Fig. 11. Total Heat Shown by Heat Diagram.

The total heat in a gas at a given pressure and volume may therefore be graphically represented by the area bounded by the

initial ordinate, the axis of X and the adiabatic curve produced to infinity. This area is

$$A=\frac{p_1v_1}{r-1}\left\{1-\left(\frac{v_1}{v_2}\right)^{r-1}\right\}_{v_2=\infty}=\frac{p_1v_1}{r-1}$$

$$\text{Total heat}=\frac{p_1v_1}{r-1}=\frac{k\,T_1}{r-1}\text{ foot pounds}=JH \tag{15}$$

$$\phi_1=\frac{H}{T_1}=\frac{k}{J\,(r-1)}$$

On the temperature-entropy diagram, the heat of the gas above 32 deg. Fahr. is represented by the area included between the ordinate, the axis of Y and the logarithmic curve of heating at constant volume. See Art. 19 and Figs. 8 and 11.

25. Change of Internal Heat.—The change of internal heat in a gas, due to any change in pressure and volume, may be graphically represented by the area included between the curve of pressure and volume and two adiabatics drawn from its extremities and produced to infinity.

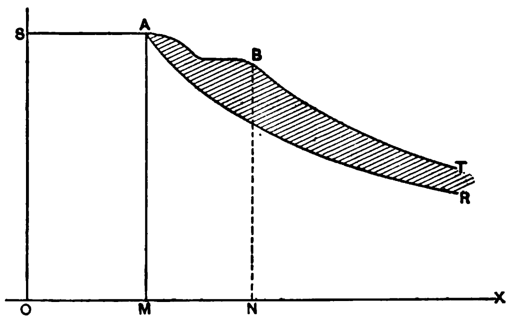


Fig. 12. Work Diagram for Change of Internal Heat.

In Fig. 12, let AB be any curve representing the change in pressures and volumes.

Draw the adiabatics AR and BT and suppose them to be infinitely extended.

Let $JH =$ initial energy of gas.

$Jh =$ final energy of gas.

$\pm JH_1 =$ heat received or rejected.

$W =$ external work done.

Then $JH + JH_1 = W + Jh$ by law of conservation of energy, or $JH_1 = W + Jh - JH$

But $JH =$ area $ARXM$

$Jh =$ area $BTXN$

$W =$ area $ABNM$

$JH_1 =$ area $ABTR$

This is much more readily shown by the temperature-entropy diagram as in Fig. 13, which is lettered to correspond to Fig. 12.

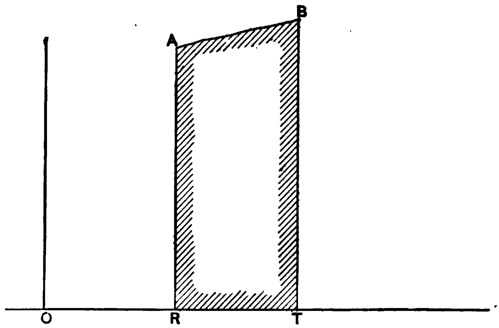


Fig. 13. Heat Diagram for Change of Internal Heat.

The area $ABTR = \int T d\phi =$ heat received.

The difference between heat received and work done, i. e. between $ABTR$ and $ABNM$, Fig. 12, is seen to be independent of shape of the curve AB and depends only on the location of the points A and B.

26. Isothermal Expansion.—Let the curve of expansion AB in Fig. 12 be isothermal; then the total heat at $A =$ area $ARXM$ $= \frac{p_1 v_1}{r - 1}$ as shown in Art. 24.

At B, the total heat=area $BTXN = \frac{p_2 v_2}{r - 1}$

But if expansion is isothermal, $p_2v_2 = p_1v_1$ and therefore the total heat or energy remains the same, as has been before stated. Consequently the heat received equals the external work done, or in figure: $ABTR = ABNM$.

To sum up: In isothermal expansion all work is done by heat received from outside, without impairing the intrinsic energy of the gas, while in adiabatic expansion all work is done at the expense of the intrinsic energy.

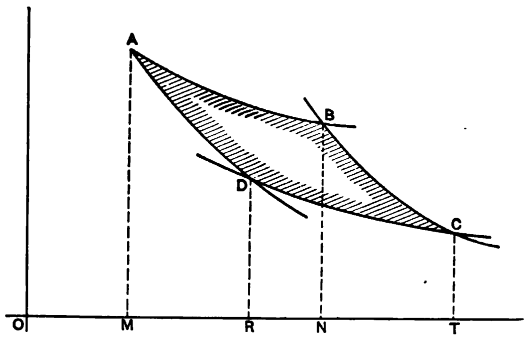


Fig. 14. Carnot Cycle—Work Diagram.

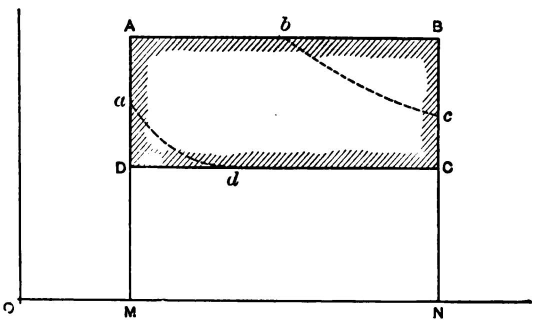


Fig. 15. Carnot Cycle—Heat Diagram.

27. Carnot Cycle.—In Figs. 14 and 15 are shown the work and heat diagrams respectively of a gas expanded and compressed in what is termed a Carnot cycle. From A to B the gas expands at a constant temperature T_1, the entropy increases from

ϕ_1 to ϕ_2 and heat is received equivalent to the rectangle $ABNM$, Fig. 15. From B to C the gas expands at a constant entropy ϕ_2 without gain or loss of heat (adiabatic) and the temperature falls to T_2.

From C to D the gas is compressed isothermally to the original entropy and heat is lost as shown by rectangle $DCNM$, Fig. 15. From D to A the gas is compressed without change of entropy (adiabatic) to the original temperature T_1. The shaded area $ABCD$ thus represents the excess of heat received or the energy converted into mechanical work.

$$H-h=(\phi_2-\phi_1)(T_1-T_2)$$

In Fig. 14, the curves AB and DC are isothermal and the curves BC and AD are adiabatics. The ordinates are pressures and the abscissas are volumes, so that the area $ABCTM$ represents positive work and the area $CDAMT$ negative work. The shaded area $ABCD$ consequently shows the net work done. As this area in Fig. 15 is expressed in heat units and in Fig. 14 is expressed in foot pounds, the numerical ratio of the second to the first will be 778, or $W=JH$ (conservation of energy).

Referring again to the so-called Carnot cycle as shown in Figs. 14 and 15, it has already been shown that the heat received during the expansion AB is represented by the area $ABNM=T_1(\phi_2-\phi_1)$, in Fig. 15; that the heat lost during the compression CD is represented by the area $CDMN=T_2(\phi_2-\phi_1)$; and finally that the shaded area $ABCD=(T_1-T_2)(\phi_2-\phi_1)$ represents the heat converted into work.

The ratio of this shaded area to the whole area $ABNM$ is then the efficiency of the engine as a machine for converting heat into work. This efficiency is evidently equal to

$$\frac{(T_1-T_2)(\phi_2-\phi_1)}{T_1(\phi_2-\phi_1)}$$

or

$$\text{Efficiency}=\frac{T_1-T_2}{T_1} \qquad (16)$$

The efficiency of a simple heat engine is then the ratio of the fall in temperature of the gas to its initial absolute temperature and this is approximately true of any heat engine.

That no engine working between the same limits of temperature and entropy can be more efficient than the one referred to is evident from Fig. 15. Let the dotted line inscribed in the rectangle represent some other law of heat change.

Then the efficiency of the process *AbcCda* is of necessity less than that of the process *ABCD*. For a maximum efficiency it is then necessary that the heat should be taken in at one temperature and rejected at another temperature and that changes of temperature and entropy should not occur simultaneously. This condition is only partially realized in the steam engine.

28. Efficiency.—The theoretical efficiency of a heat engine obtained by dividing the work done by the total heat in the gas and which can never exceed $\frac{T_1 - T_2}{T_1}$ is hardly a fair standard.

The natural environment of actual engines makes it entirely impracticable to use any considerable fraction of the total heat. It is as unfair to use this total for a divisor as it would be to establish the zero for a water power at the center of the earth, and to calculate the efficiency of the fall by dividing the available head by the radius of the earth. We may, however, consider the shaded area in Fig. 15, $(T_1 - T_2)\ (\phi_2 - \phi_1)$, as a standard of performance for air engines working between these limits and compare the actual work done with this standard.

A different standard is used for steam engines as will be explained in the next chapter.

29. The Air Compressor.—Compressed air is frequently used for driving portable machines at some distance from the source of power, such as rock drills in mines and riveting and calking machinery in shipyards. The air is taken at the normal pressure and temperature, compressed to any desired degree and then conveyed to the machine in pipes or hose.

When air is compressed, stored and finally expanded in doing work, the only heat cycle is that due to the losses by cooling. Referring to Fig. 16, let air at pressure and volume *A* be compressed adiabatically to pressure and volume *B*, the air will be heated as shown in Art. 12. If the air could be kept hot until it was used in the engine it would expand down the adiabatic *BA* to

the original pressure, volume and temperature and no heat would be lost. The usual cycle, however, is this: compression with rise of temperature along *AB;* cooling at constant pressure along *BC* to normal temperature; expanding and doing work from *C* to *D* with falling temperature; warming at atmospheric pressure to original condition at *A*.

The area *ABCD* thus represents the loss due to radiation and conduction.

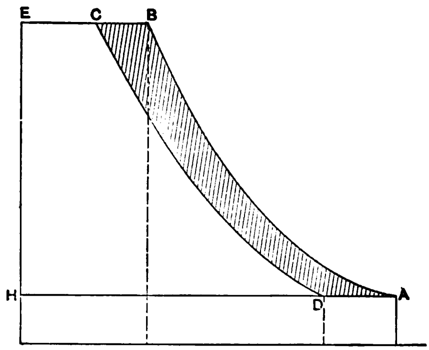


Fig. 16. Air-Compressor Diagram.

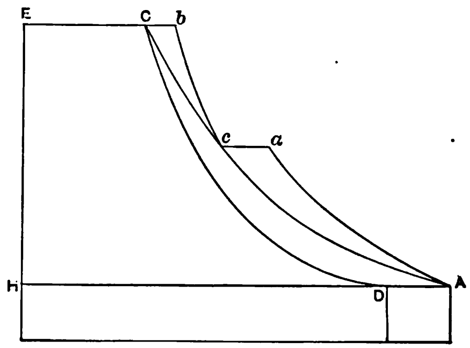


Fig. 17. Stage Compression.

If it were possible to compress the air isothermally along *AC*, Fig. 17, rejecting a certain amount of heat, and then utilize this

same heat to give isothermal expansion, no heat would be lost, the compression and expansion being on the same line.

An approach to this is made by compressing the air in one cylinder to some point as *a*, cooling it to *c*, and finally compressing again to *b* in a second cylinder, Fig. 17.

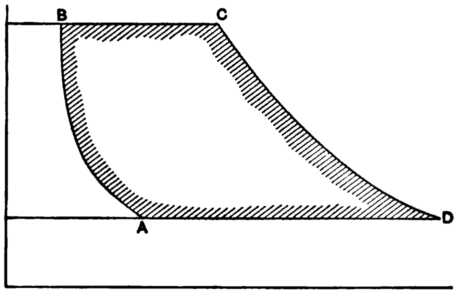


Fig. 18. Hot-Air Engine Diagram.

If several cylinders were used in this way it is evident that the broken line would approach still closer to the isothermal *AC* and the heat loss be further reduced. In a similar manner two or three stages of expansion might be used, but this would involve a supply of heat from outside the system.

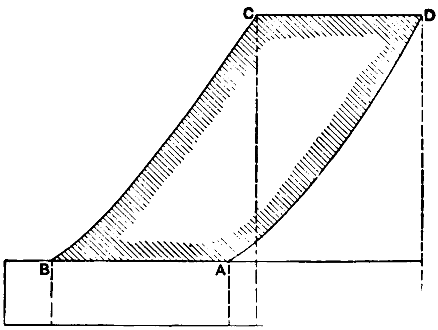


Fig. 19. Entropy Diagram Corresponding to Fig. 18.

30. Hot Air Engines.—In hot air engines of the modern type the air is heated by a furnace and cooled by a water jacket or by coils. The action of the air can be understood from the dia-

gram, Fig. 18, without considering the mechanism employed. Starting with air of low pressure and temperature at *A*, a compressor piston compresses the air to *B* isothermally since it is in contact with cold surfaces.

Leaving this cylinder the air passes through a hot regenerator to a larger working cylinder, expanding by heat to the volume *C*. Cut-off takes place and the air expands isothermally in the large cylinder to *D*, being in contact with hot surfaces. *DA* shows the rejection of the air at a constant pressure.

Fig. 19 is the corresponding entropy diagram with lettering arranged in the same order.

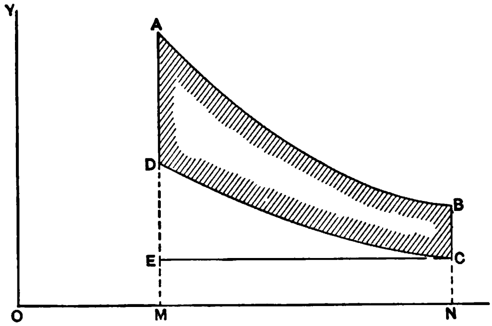


Fig. 20. Otto Cycle.

From *A* to *B* heat is rejected at a constant temperature; from *B* to *C* is the line for heating at a constant pressure (Art. 20, Fig. 8); from *C* to *D* heat is received at a constant temperature and from *D* to *A* the air is cooling at a constant pressure. The shaded area represents the net heat received and converted into work.

The efficiency of the hot-air engine is good, but it is very bulky in proportion to its power. Small engines of this class are much used for domestic pumping service.

31. Gas Engine Cycles.—The gas engine is simply a hot-air engine in which the air is heated by the explosion of gas inside the cylinder of the engine itself.

The diagrams are interesting from a thermodynamic standpoint and vary with the type of engine used. For illustration we will use the so-called Otto cycle, the diagram of the well-known Otto engine. Fig. 20 shows the cycle of operations which is as follows:

(1) The piston moves forward drawing in the explosive mixture through the admission valve at a constant pressure (line *EC*).

(2) The piston returns, compressing the mixture into the large clearance space *OM*, the admission valve being closed. (Line *CD*.)

(3) The mixture is ignited and explodes with a sudden increase of temperature and pressure. (Line *DA*.)

(4) The hot mixture expands along a curve *AB* which is nearly adiabatic.

(5) The exhaust valve opens and the pressure suddenly falls (line *BC*) when the mixture is expelled from the cylinder at a constant pressure. (Line *CE*.)

Let

$$OM = v_1 \qquad BN = p_2$$
$$ON = v_2 \qquad CN = p_3$$
$$AM = p_1 \qquad DM = p_4$$

and let *AB* and *CD* be adiabatics for dry air.

Suppose for convenience that the engine uses 1 lb. of the mixture per stroke. The work done is shown by the shaded area *ABCD*.

$$\text{Area } ABNM = \frac{p_1 v_1}{n-1}\left\{1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right\}$$

See Art. 12.

$$\text{Area } CDMN = \frac{p_4 v_1}{n-1}\left\{1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right\}$$

$$\text{Area } ABCD = (p_1 - p_4)\frac{v_1}{n-1}\left\{1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right\}$$

But
$$p_1 = p_4 \times \frac{T_1}{T_4}$$

as the volume is constant during the heating, and

$$p_1-p_4=p_4\frac{T_1-T_4}{T_4}$$

Substituting this value of (p_1-p_4) in the equation for work,

$$W=\frac{p_4v_1}{n-1}\left\{1-\left(\frac{v_1}{v_2}\right)^{n-1}\right\}\frac{T_1-T_4}{T_4} \tag{17}$$

or the useful work done is proportional to the rise in temperature.

If we assume the specific heat of the mixture to be the same as that of air, the heat supplied in raising the temperature will be

$$JH=131.5(T_1-T_4)\text{ ft. lb.}$$

(See Art. 18.)

$$\text{The efficiency}=\frac{W}{JH}=\frac{\frac{p_4v_1}{n-1}\left\{1-\left(\frac{v_1}{v_2}\right)^{n-1}\right\}}{131.5\,T_4}$$

But by Art. 24

$$\frac{p_4v_1}{n-1}=131.5\,T_4$$

= heat in the gas at T_4, i.e. $\frac{kT}{n-1}=C_vT.$

Therefore the efficiency of the cycle is:

$$E=1-\left\{\frac{v_1}{v_2}\right\}^{n-1}=1-\frac{T_2}{T_1} \tag{18}$$

As may be seen from the figure, the heat given to the gas at DA is represented by the area between the curves AB and DC extended to infinity, consequently if the expansion were infinite or $\frac{v_1}{v_2}=0$, all the heat would be converted into useful work.

The entropy diagram for the Otto cycle shows the difference between the efficiency of this and of the Carnot cycle. In Fig. 21 the lines AB and CD are the constant entropy lines due to a non-conducting cylinder, while BC and DA show the cooling and heating at constant volume (see Art. 19). The heat received is area $MDAN=C_v\ (T_1-T_4)$ and the heat rejected is area

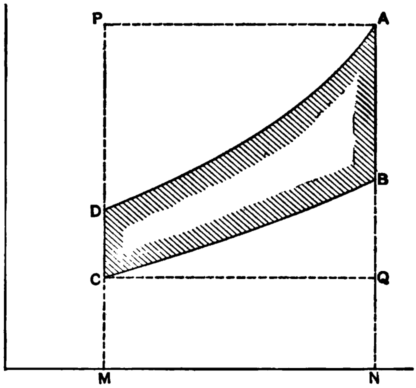


Fig. 21. Entropy Diagram for Otto Cycle.

$NBCM = C_v\ (T_2 - T_3)$. The heat converted into work is the difference shown by the shaded area and $= C_v\ (T_1 - T_2 + T_3 - T_4)$. The efficiency therefore is

$$\frac{T_1 - T_2 + T_3 - T_4}{T_1 - T_4} = 1 - \frac{T_2 - T_3}{T_1 - T_4}$$

But as the change of entropy is the same in both heating and cooling

$$\phi_1 - \phi_4 = C_v \log_e \frac{T_1}{T_4} = C_v \log_e \frac{T_2}{T_3}$$

and $\quad \dfrac{T_1}{T_4} = \dfrac{T_2}{T_3} \quad$ or $\quad \dfrac{T_2}{T_1} = \dfrac{T_3}{T_4}$

$$\text{efficiency} = 1 - \frac{T_2}{T_1} \text{ as before.}$$

But for the Carnot cycle shown in dotted lines

$$\text{efficiency} = 1 - \frac{T_3}{T_1}$$

or considerably more than the other.

The above is what is commonly called a four-cycle engine.

In the two-cycle engine there is an explosion at each revolution and the return stroke is employed in introducing the fresh charge and driving out the burnt gases at the same time.

The practical difficulty in accomplishing this has made two-cycle engines more or less unsatisfactory. The heat-process and the heat-diagram is the same as for the four-cycle engine.

32. **Actual Efficiencies.**—As might be expected the efficiency of the real gas engine is much less than that of the ideal on account of the loss of heat by radiation and conduction.

In practice it is found necessary to surround the cylinder with a water-jacket to carry off the heat which would otherwise be stored in the metal of the walls and raise the temperature to a dangerous degree.

Various experiments on Otto engines of from 10 to 15 h.p. show average working efficiencies about as follows:

Useful work	15 per cent.
Friction of engine	5 per cent.
Loss at exhaust	25 per cent.
Heating water-jacket	45 per cent.
Radiation	10 per cent.
	100

More heat may go out of the exhaust and less to the jacket but the amount converted into work is rarely over 20 per cent.

An analysis of a test on a 10 h.p. gasolene engine is reported in *Power* for May, 1903, and shows the following distribution of work in percentages:

Useful work	19.10
Friction of engine	3.70
Loss at exhaust	32.70
Heating water-jacket	38.50
Radiation	6.
	100.

The proportion of gas to air by weight is as one to seven for maximum efficiency, according to experiments, and the consumption of gas may be from 20 to 25 cu. ft. per indicated horse power per hour.

The value of n depends upon circumstances, but is usually from 1.3 to 1.4, the water-jacket modifying the shape of the expansion and compression curves.

In actual diagrams the line *DA,* Fig. 20, is inclined to the right, the explosion not being instantaneous, and the toe of the diagram is rounded by wire-drawing.

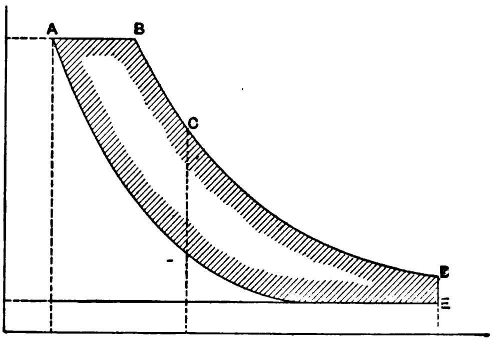


Fig. 22. Diesel Engine Diagram.

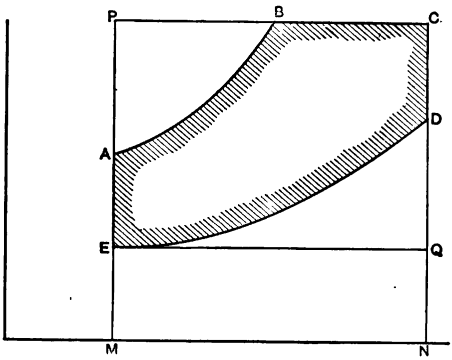


Fig. 23. Entropy Diagram Corresponding to Fig. 22.

33. The Diesel Engine.—The Diesel oil engine is the result of an attempt to realize the high efficiency of the Carnot cycle. The success of this attempt may be judged by a comparison of diagrams.

The practical operation of the engine is as follows: Air is drawn in on the initial stroke and on the next stroke is compressed

adiabatically to a high pressure, about 500 lb. per square inch. This produces such a high temperature that when oil is pumped into the cylinder by an auxiliary pump, combustion at once begins.

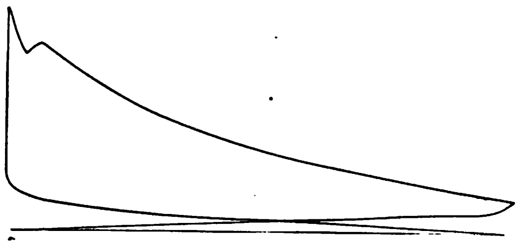


Fig. 24. Indicator Diagram, Four-Cycle Engine.

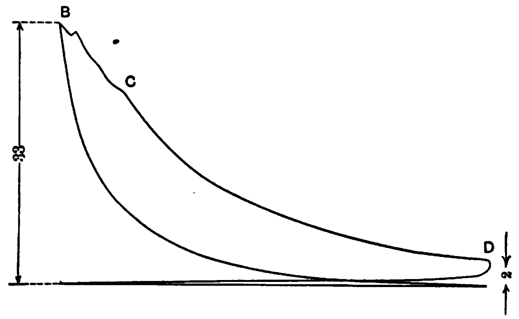


Fig. 25. Indicator Diagram, Diesel Engine.

This combustion produces a further increase in temperature, but the oil is then pumped in just fast enough to maintain isothermal expansion until cut-off. After cut-off the expansion is adiabatic or nearly so, down to the terminal pressure.

The rejection of the burnt gases is the same as in the Otto engine.

Figs. 22 and 23 show the pressure-volume and the temperature-entropy diagrams for this engine, similar letters showing corresponding points in the two.

The operations are as follows:

EA, compression of air accompanied by rise of temperature.

AB, injection of oil at constant pressure causing further rise of temperature.

BC, isothermal expansion due to further combustion.

CD, adiabatic expansion after cut-off.

DE, cooling at constant volume.

A comparison of Fig. 23 with Fig. 21 shows the advantage of the Diesel cycle to consist in the gradual combustion at a constant temperature shown by *BC* which makes the cycle approach more closely to the Carnot cycle *EPCQ.*

For the same temperature range the Diesel cycle has therefore a higher efficiency than the Otto.

Actual tests of this engine show efficiency to range from 23 per cent at half load to 26.6 and 28.6 per cent at full load. (Compare with Art. 32.)

Figs. 24 and 25 are from indicator diagrams of Otto and Diesel engines respectively. Various other gas and air engine cycles, such as the Lenoir, the Joule and Stirling are discussed in treatises on thermodynamics, and are interesting from a scientific standpoint. As they do not represent the work of successful modern engines they are not introduced here.

Problems.

1. Five cubic feet of air at atmospheric pressure and a temperature of 72 deg. Fahr. are heated to 184 deg. Fahr. without change of volume.

Find the weight of air, the resultant pressure and the heat required (a) in heat units, (b) in foot pounds.

2. If the air in (1) is heated at a constant pressure, find the final volume and the heat required.

3. Find the change of entropy in Problems 1 and 2 and draw a diagram to illustrate.

4. Find the heat in the air above 32 deg. Fahr. in Problem 1.

5. One pound of air at a pressure of 85 lb. per square inch and a temperature of 68 deg. Fahr. is expanded isothermally to four times the original volume.

Find the change of entropy and the heat received. What is the work of expansion?

6. If the air in Problem 5 is expanded in a non-conducting cylinder, what will be the final temperature?

7. In Problem 6 what fraction of the total heat in the air is expended in work?

8. Let 1 cu. ft. of dry air at an initial temperature of 70 deg. Fahr. be expanded and compressed in a Carnot cycle and let

$$\frac{v_2}{v_1} = 2 \quad \frac{v_3}{v_2} = 3$$

(See Figs. 14 and 15.)

Determine:

(1) The lowest temperature.

(2) The change of entropy *AB*.

(3) The heat received and rejected and the efficiency.

9. In a cycle like that shown in Fig. 16, find the work of compression and that of expansion in foot pounds, the heat lost in thermal units, and the efficiency of the operation.

Assume 1 lb. of dry air to be used at 60 deg. Fahr. and the initial and final pressures to be 15 lb. and 105 lb. absolute. State volumes and temperatures at *A, B, C* and *D*.

10. Work out the same problem for the cycle shown in Fig. 17, assuming that the air is cooled to original temperature at 60 lb. pressure.

11. Draw to scale an entropy diagram for the cycle in Fig. 16.

12. If the hot-air engine, whose cycle is shown in Fig. 18, works between the pressures of 15 lb. and 105 lb. and the temperatures 100 and 500 deg. Fahr., find the foot pounds of work developed per pound of dry air.

13. Find the efficiency of the cycle from Fig. 19, using the same data as in Example 12.

14. A gas engine working on an Otto cycle like that in Fig. 20, shows the following gage pressures on the indicator card: $A = 270$ lb., $B = 42$ lb., $C = 0$ lb., $D = 60$ lb.

If the expansion and compression are adiabatic and the gas enters at 60 deg. Fahr., find the temperatures at *A, B* and *D*. (See diagram, Fig. 24.)

15. Find the efficiency of the cycle in preceding example and compare with efficiency of Carnot cycle under similar conditions.

16. Give the equation of each of the lines in Fig. 23.

CHAPTER IV.

THE THERMODYNAMICS OF STEAM.

34. The Formation of Steam.—In converting water into steam, heat is expended in three ways: (1) Heating the water to the temperature of evaporation, commonly called the heat of the liquid—internal work. (2) Changing the water into steam—internal work of separating the molecules. (3) Expanding the volume from that of water to that of steam—external work against whatever may be the pressure.

The last two together are commonly termed the latent heat of evaporation or simply the latent heat. The sum of the three is called the total heat of the steam at that pressure and is commonly reckoned from 32 deg. Fahr. This is shown graphically in Fig. 3.

Let

$q =$ heat of the liquid per lb.
$L =$ latent heat per lb.
$H =$ total heat per lb.
$t =$ temperature of evaporation in deg. Fahr.

Then from experiments of Regnault,

$$H = 1091.7 + 0.305(t-32)$$

$$H - L = q = t - 32 \text{ nearly}$$

$$L = H - (t-32) = 1091.7 - 0.695(t-32), \text{ or approximately,}$$
$$L = 1092 - 0.7(t-32)$$
$$= 966 - 0.7(t-212)$$

The relations between the temperature of boiling and the pressure are determined by experiment and may be found in the first two columns of the steam tables. The heat of the liquid q may be obtained approximately by subtracting 32 deg. from the temperature of boiling. The specific heat of water is nearly constant, but increases slightly as the temperature rises, so that q is always greater than $t_2 - t_1$.

The total heat is the sum of the heat of liquid and the latent heat or $H = q + L$. The relations between the temperature and the specific volume are also experimental. The volume and pressure of superheated steam obey the law of permanent gases, $pv = kT$, where k is obtained from the values of p, v and T at saturation.

35. The Entropy of Steam.—As the heat in saturated steam consists of two parts, the heat q, acquired during the rise of temperature of the water, and the heat L acquired at a constant temperature during evaporation, so may the entropy be regarded as consisting of two parts, the entropy of the water added during the first stage and the entropy of the steam added during the second stage.

The first is commonly denoted by θ and the second by ϕ.

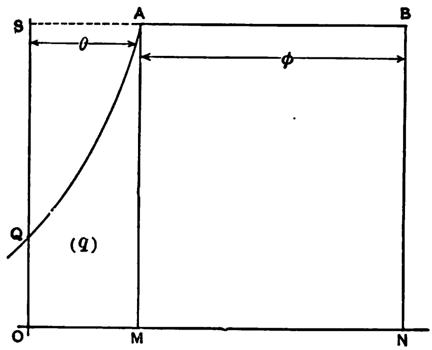


Fig. 26. Entropy of Steam and Water.

The latter is easily calculated from the steam tables by dividing the heat of evaporation by the absolute temperature, or in symbols

$$\phi = \frac{L}{T}$$

This is illustrated in Fig. 26, where MA represents any absolute temperature of boiling T, and the rectangle $ABNM$ the heat of evaporation L received at a constant temperature. Then AB

represents the entropy of the steam at that temperature. For instance, at a pressure of 30 lb., the temperature of boiling is

$$T = 460.7 + 250.3 = 711$$

and

$$L = 938.9$$

The entropy of steam is then

$$\phi = \frac{L}{T} = \frac{938.9}{711} = 1.32.$$

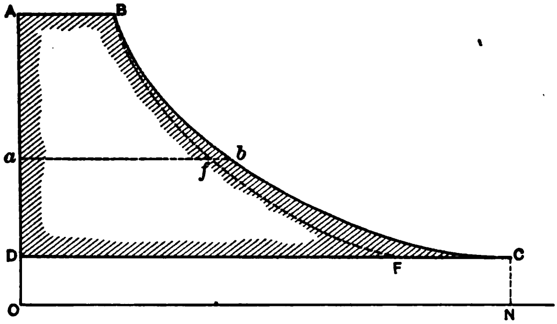


Fig. 27. Cycle of Ideal Steam Engine.

36. **The Entropy of Water.**—This is illustrated in Fig. 26, where the line QA represents the equation

$$\theta - \theta_0 = c_v \log_e \frac{T}{T_0} \quad \text{(See Art. 19.)}$$

If $OQ = T_0$ is taken as 492 deg. then $\theta_0 = 0$ and the area $QAMO$ represents q, the heat of liquid above 32 deg. Fahr., while OM represents θ, the entropy of water for the given temperature.

Using the same examples as in the preceding article and calling $c_v = 1$.

$$T = 711 \text{ and } \theta = \log_e \frac{711}{492} = \log_e 1.443 = 0.368.$$

37. **The Saturated Steam Curve.**—If a curve is plotted by using pressures for ordinates and the corresponding specific vol-

umes of saturated steam for abscissas, this will represent the expansion of dry saturated steam which receives just enough heat from outside to prevent condensation. Such a curve is shown in BC, Fig. 27.

The corresponding curve on the $T\phi$ diagram is plotted by using for ordinates the absolute temperatures of the steam and for abscissas the sums of the corresponding entropies of water and steam.

Thus, referring to the examples worked out in Arts. 35 and 36, the coördinates for steam at 30 lb. pressure would be

$$T=711$$

$$\theta+\phi=1.688.$$

The line BC in Fig. 28 is such a curve and corresponds to the line BC on the pv diagram, Fig. 27.

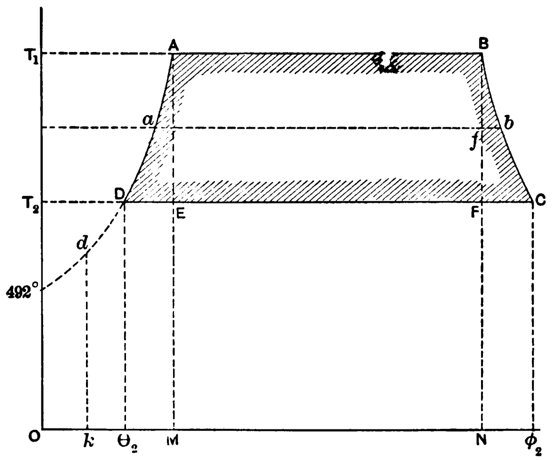


Fig. 28. Temperature-Entropy Diagram for Steam.

38. The Adiabatic Curve.—If a vertical line BN be drawn in Fig. 28 it will represent fall of temperature without flow of heat, such as occurs during adiabatic expansion.

If horizontal temperature lines, ab, DC, be drawn, their lengths will represent the entropy of the steam at these temperatures if

dry and saturated. If the entropy of the steam at the temperature OT_2 is only DF it is because a portion corresponding to FC has condensed and the fraction $\frac{FC}{DC}$ is the percentage of moisture present. In a similar manner, the moisture fraction at ab is $\frac{fb}{ab}$. All points to the left of BC represent wet steam and all points to the right of BC represent superheated steam.

To obtain the pv curve for adiabatic expansion in Fig. 27, it is only necessary to draw the corresponding pressure lines, ab, DC, on that figure and lay off the distances bf, CF, etc., so that the ratios $\frac{fb}{ab}$ and $\frac{FC}{DC}$ may be the same as in Fig. 28. A sufficient number of such points will serve to locate the adiabatic curve BF. The equation of the curve is very nearly $pv =$ a constant.

39. The Cycle for Steam.—The diagram shown in Fig. 27 gives the cycle of events in the cylinder of an ideal engine.

From D to A steam is admitted, the temperature and the pressure increasing; from A to B steam is admitted at a constant temperature T_1 and pressure p_1; from B to C the steam expands with the temperature and pressure falling, receiving just enough heat to prevent condensation; i.e. BC is the curve for saturated steam corresponding to the values for p, t and d given in the steam tables; from C to D steam is expelled at a constant temperature T_2 and pressure p_2; the mechanical work done is now represented by the shaded area.

If no heat is received by the steam, the expansion line is BF and a part of the steam is condensed.

The diagram $DABF$ is called the Rankine cycle and is used for purposes of comparison with actual steam diagrams. Its area is

$$A = p_1v_1 - p_2v_2 + \frac{p_1v_1 - p_2v_2}{r - 1}$$

$$= \frac{r}{r-1}(p_1v_1 - p_2v_2) \qquad (18)$$

Fig. 28 shows the corresponding heat diagram and the changes of entropy and temperature. From D to A heat is flowing into the water and its temperature rises to the boiling point.

The area under the curve represents the heat received by the water, $q_1 - q_2$.

From A to B the water receives heat L_1 at a constant temperature T as it evaporates and at B is dry saturated steam. If it remains dry while expanding, heat must be received from some outside source equal to the area under the curve BC. (The lines DA and BC are nearly straight and are sometimes so drawn as a sufficiently close approximation.)

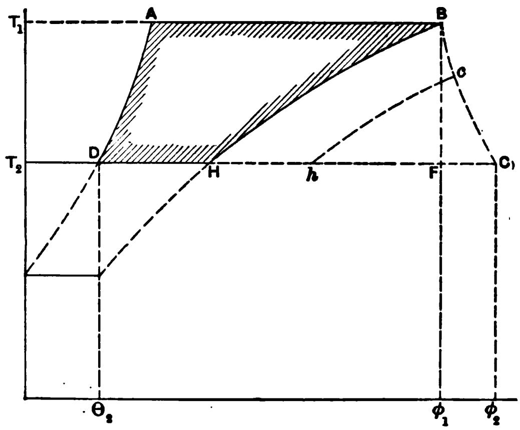


Fig. 29. Constant Volume Lines.

From C to D the steam is condensed or rejected from the cylinder and the area under the line is the heat of evaporation L_2 at the lower temperature. This area represents loss of heat and the shaded area the heat converted into mechanical work.

40. Diagram Without Expansion.—If in Fig. 27 the steam instead of expanding along the line BC is allowed to escape from the cylinder at B the resulting fall of pressure would be shown by a vertical line through B. The corresponding line BH on the entropy diagram, Fig. 29, is called a constant volume line and may be drawn in a manner similar to that already described for the line BF in Figs. 27 and 28. It may also be drawn directly by calculation from the steam tables. For instance, if the temperature

at B is 350 deg. Fahr. and that at C is 230 deg. Fahr. the table shows the volumes of 1 lb. of steam to be 3.32 and 19.01 cu. ft. respectively.

Accordingly, if the volume remains constant at 3.32 cu. ft. the final weight of steam will only be

$$\frac{3.32}{19.01}$$

or 0.175 the initial weight. The entropy DH will accordingly be only 0.175 of DC, the entropy of 1 lb. at the lower temperature.

This is the case of steam working without expansion and the ratio

$$\frac{\text{work done}}{\text{heat received}} = \frac{ABHD}{\theta_2 DAB\phi_1}$$

shows the low efficiency of such an arrangement. Similar constant volume lines as ch, Fig. 29, may be drawn from different points on the dry steam line BC to represent the effect of more or less complete expansion.

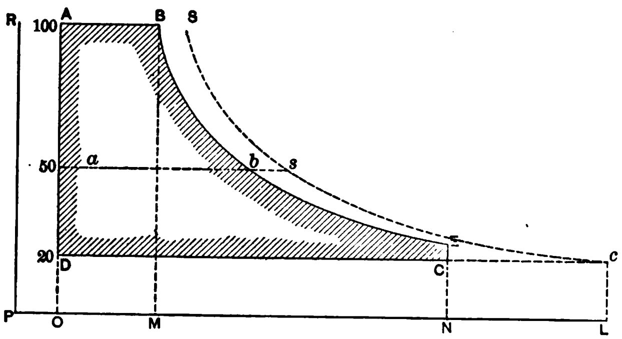


Fig. 30. Expansion of Wet Steam.

41. Wet Steam.—The steam in an engine cylinder is usually wet at cut-off on account of initial condensation. The steam from the boiler, at a temperature of over 300 deg. Fahr., comes in contact with the comparatively cool surfaces of the cylinder and piston heads which have just been exposed to the exhaust

and a considerable percentage is condensed and deposited as a film of water on those surfaces. This condensing process goes on through the stroke as fresh surfaces of cool metal are uncovered by the piston, but as the temperature and pressure of the expanding steam fall and the cylinder grows warmer, some of the water deposited at the first of the stroke is evaporated again.

The actual amount of dry steam present at any point in the stroke can be determined by inspection of the indicator diagram. Usually with an early cut-off more dry steam is found present at release than at cut-off.

In Fig. 30 let $ABECD$ be the indicator diagram of an engine taking steam at 100 lb. absolute and exhausting at 20 lb. absolute. We will neglect the effects of wire drawing, compression and clearance.

Let AS be the volume which the steam would occupy at cut-off if dry, and let Ssc be the curve of saturation.

The quality of the steam at cut-off is $\frac{AB}{AS} = 75$ per cent dry and at b the quality is $\frac{ab}{as}$ or about 80 per cent dry, showing that the evaporation since cut-off has more than balanced the condensation.

In Fig. 31 draw the horizontal lines ABS and DCc corresponding to the absolute temperatures of admission and exhaust or 788 deg. and 688 deg. respectively.

The part of the figure below 600 deg. absolute is omitted to save room. The entropies of water at these two temperatures are 0.473 and 0.336. Locate the points A and D accordingly.

Lay off $AS = 1.12$, the entropy of dry steam at 788 deg. absolute and $Dc = 1.384$, the entropy of steam at 688 deg.

The temperature-entropy diagram $AScD$ corresponds to the complete expansion of dry steam as shown by the dotted lines in Fig. 30.

But, as explained in Art. 38, the percentage of entropy in the heat diagram corresponds to the percentage of dry steam present. Draw horizontal lines as abs, Fig. 31, to represent temperatures corresponding to the various pressures in Fig. 30.

Locate the points B, b, E, C, etc., so that the ratios $\frac{AB}{AS}, \frac{DC}{Dc}$ etc., may equal those in Fig. 30 and plot the curves BE and EC.

Then will the shaded area $ABECD$ represent the heat converted into work per pound of steam delivered by boiler.

The total heat received from the boiler will be represented by the area under $DABS$ extending to the line of absolute zero, plus heat used in heating feed water to 688 deg. The heat lost by condensation before cut-off is shown by the rectangle under BS, but a part of this is restored by the cylinder walls during expansion BE. It will be understood that the point E may fall to the left or right of the vertical line through S. The efficiency is visibly less than if the steam had remained dry as per dotted lines.

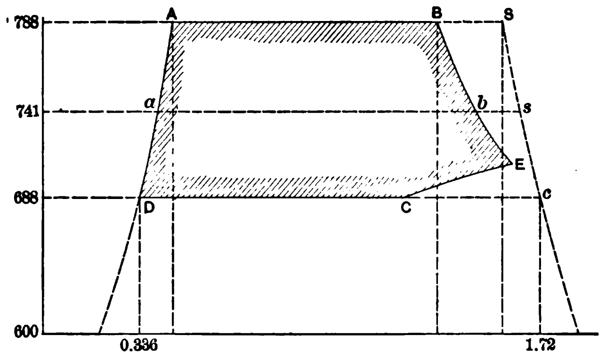


Fig. 31. Heat Diagram Corresponding to Fig. 30.

42. Superheating.—The heating of steam after it has been removed from contact with the water raises its temperature to a point above that due to its pressure and the more it is thus superheated, the further is it removed from the critical point and the more it resembles a gas.

Such steam may accordingly be cooled down to the temperature of saturation without condensation, and this makes it a desirable medium for use in steam engine cylinders.

Fig. 32 shows the entropy diagram for steam which is superheated before expansion. As before, DA represents the heating of the water and AB the evaporation. The curved line BRS

shows the change of temperature and entropy due to superheating at a constant pressure, and the area under the line the amount of heat used in the process. Like *DA* the line *BS* is a logarithmic curve. (See Art. 36.)

Until recently the value 0.48 has been accepted for the specific heat C_s of superheated steam.

The results of the more recent investigations of this subject are given in Chapter X.

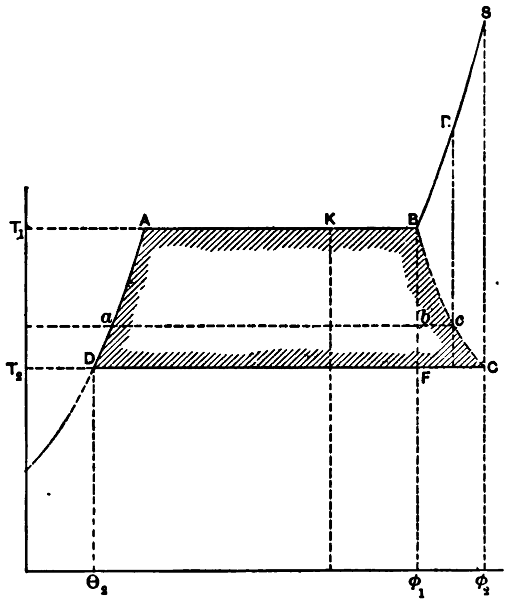


Fig. 32. Heat Diagram for Superheated Steam.

If the steam expand adiabatically from *S* it will gradually lose its superheat until when it reaches the dry steam line at *C* it becomes again saturated steam.

If there is a less degree of superheat as at *R* and the steam expands as before, it becomes dry steam at *c*, and if the expansion continues some of the steam will condense.

Superheated steam on entering an engine cylinder usually cools at approximately a constant pressure and the line *BRS* is retraced.

If there is just enough heat to keep the steam dry the cooling stops at B and expansion begins.

Less superheat will allow the steam to condense along BK and expansion will begin at K, giving a diagram similar to Fig. 31.

The principal advantage of using superheated steam is the prevention of this initial condensation in the engine cylinder.

43. Steam Consumption.—The actual amount of steam used by an engine may be determined by passing the steam through a surface condenser and then weighing it in a tank on scales. The amount of dry steam present at different parts of the stroke may be determined by measuring the indicator diagram as in Fig. 30. Let it be assumed that the steam is saturated during expansion, as there is always water present unless superheat is used. Then the temperature and density at any point as B or E, Fig. 30, may be determined from the steam table. From this the weight of dry steam present may be calculated. Referring again to Fig. 30 for illustration, the volume of dry steam present at B is represented by the distance RB from the clearance line. The ratio $\frac{RB}{DC}$ times the piston displacement in cubic feet, gives the number of cubic feet represented by the distance RB. If the pressure is 100 lb., the weight of a cubic foot of steam is 0.227 lb. The product of this by the volume at B gives the weight of dry steam present. A similar calculation may be made for any other point as "b," using in this case the weight of a cubic foot of steam at 50 lb.

Let $W =$ actual weight of steam used per stroke as determined by a condenser and weighing tank.

Let $x =$ indicated weight of steam at cut-off as determined from point B in Fig. 30.

Let $y =$ indicated weight of steam at release or point E in Fig. 30.

Let $z =$ indicated weight of steam at any point on compression curve (not shown in Fig. 30).

The steam is usually assumed to be dry during compression since all the water will be evaporated during the low pressure interval between C and the beginning of compression. This weight of steam is always present in the cylinder and is not in-

cluded in the amount W which passes through. The total weight of the mixture to be accounted for at any time during expansion is then $W + z$. By comparing the amount of steam shown by indicator at any point in the expansion line with this total amount of the mixture present, the percentage of dry steam can readily be obtained, or in other words,

$$\frac{x}{W+z} = \text{per cent of dry steam at cut-off.}$$

$$\frac{y}{W+z} = \text{per cent of dry steam at release.}$$

The latter ratio will usually be larger on account of reëvaporation during expansion. In making these calculations allowance should be made for any initial moisture in the steam as shown by a calorimeter in the steam pipe.

44. Factors of Evaporation.—The true unit for all problems of thermodynamics is the heat unit, but it is sometimes more convenient from a practical standpoint to measure the amount of heat used in pounds of steam. It is necessary in such cases to specify the pressure and temperature of the steam as these determine its heat value.

In order to get a common standard of comparison it is customary to express steam measurements in terms of dry saturated steam at atmospheric pressure, produced from water of the same temperature or, in common parlance, steam from and at 212 deg. The heat required to produce such steam is 965.8 B.t.u.

To reduce the actual amount of steam used in any case to terms of the equivalent steam from and at 212 deg., we have to divide the heat used per pound of steam by 965.8 and obtain the so-called *factor of evaporation.*

Multiplying the quantity of steam used by this factor will give the equivalent in the new units. Assume, for example, that the steam is dry and saturated at 105 lb. pressure and the temperature of the feed water is 164 deg. The total heat of the steam above 32 deg. Fahr. is by the tables $H = 1181.9$. Subtracting from this the heat in the feed water above 32 deg. and we have

$$1181.9—(164—32) = 1049.9.$$

The factor of evaporation is accordingly:

$$\frac{1049.9}{965.8}=1.087$$

and the number of pounds of steam made must be multiplied by this factor to reduce to terms of the equivalent steam from and at 212 deg. If the steam is superheated, the heat due to that condition must be added in determining the total heat.

The factor of evaporation is more used in connection with boiler tests than in engine testing.

45. Refrigeration Cycles.—The object of a refrigerating plant is to abstract heat from some substance and thereby lower its temperature below that of surrounding bodies. Such a plant may be used to cool water, air or some other substance, but the principle in all cases is the same, and the process is the reverse of that used in the various heat engines which have been discussed in this book.

The object of a heat engine is to put heat into a gas or vapor and convert this into mechanical work. The object of a refrigerating machine is to draw heat from a substance by the exertion of mechanical energy. In the first case the object is to make the mechanical work as large as possible, in the second to keep it as small as possible.

In either case the gas or vapor is used merely as a medium of exchange, as a heat carrier, and the object of the process is to effect an interchange between heat and work. Air, being plentiful and cheap, has been tried as a medium in both processes, but neither air engines nor air refrigeration machines are convenient practically, on account of the large size of cylinders required and the great extremes of temperature which occur.

For a heat engine a medium is desired which has a boiling point above ordinary atmospheric temperatures and can be handled as a liquid. Water serves admirably for this purpose.

For refrigerating purposes a medium is best which has a boiling point below the normal temperature of the atmosphere and which under ordinary circumstances is a vapor. Only two such substances are in common use, ammonia and carbon dioxide.

Probably 95 per cent of refrigerating machines are of the ammonia type and only these will be considered here.

In order to understand the difference between the steam engine cycle and that of the ice machine, it will be well to study an ideal diagram of the two processes, as shown by Professor Hutton in his *Heat and Heat Engines.*

In Fig. 33 let the working fluid circulate through the pipes and reservoirs shown. Let B represent a receptacle which heats the fluid and C one which cools it. Let P represent a pump which carries the fluid from B to C or contra, and E a cylinder where

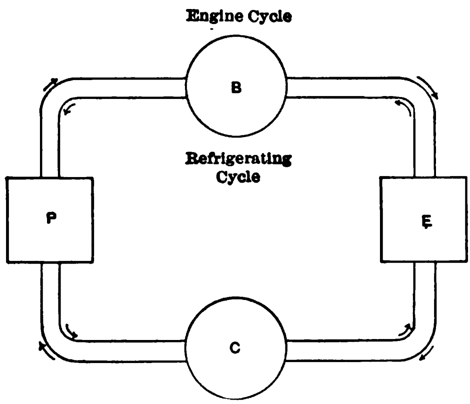


Fig. 33. Comparison of Engine and Refrigerating Cycles.

expansion takes place. If the machine is a heat-engine then the circulation will be clock-wise; B will be the boiler where heat changes the water to steam; E the engine where expansion of the steam converts heat into mechanical work, C the condenser where heat is abstracted from the steam and P the feed pump carrying water from the hot-well to the boiler.

The object of the cycle is to do work at E and this is done by means of heat received at B.

If the machine is for refrigeration the cycle is reversed and becomes contra-clock-wise. The warm ammonia-vapor at a high pressure goes from the compressor at P to the cooler C, where it is condensed to a liquid; E is the expansion valve where the

liquid is allowed to vaporize and expand against the lower pressure beyond. *B* is the brine tank where the now cold vapor receives heat from the brine surrounding the coils, thereby cooling the latter, and finally *P* is the compressor which draws the vapor from *B* and compresses it into *C*.

The object of this cycle is to draw heat from *B* and the energy by which this is done comes from *P*.

From *P* around through *C* to *E* there is high pressure and from *E* around through *B* to *P* there is low pressure.

To pump from the low pressure to the high is the function of the compressor. Pure anhydrous ammonia is the fluid used as a heat-carrier. The characteristics of this substance are given in Table XII., which was originally calculated by Prof. DeVolson Wood.

As ammonia is a comparatively expensive material it is used over and over in the process and not rejected from the system, as is the steam from an engine. The line from *E* through *B* to *P* is the suction line, having a pressure varying from 5 to 20 lb. gage. This pressure depends upon the speed of the compressor and the opening of the expansion valve. For a temperature of 0 deg. Fahr. in the cold storage at *B*, a pressure of about 5 lb. gage is maintained, while a pressure of 20 lb. corresponds to a temperature of about 32 deg. Fahr., there being usually 12 or 15 deg. difference between the temperature of the ammonia and that of the room cooled.

The ammonia vapor is compressed at *P* by a compressor mechanically driven and is delivered to *C* at a pressure of from 150 to 200 lb. gage and a temperature of from 70 to 100 deg. Fahr.

The vapor at this stage may be saturated or superheated. If liquid ammonia is present in the compressor the process is called wet and the temperatures will be limited to that corresponding to the pressure. On the other hand the vapor, if dry, will become superheated and the temperatures will range much higher, as in the so-called dry process. At *C* the vapor is cooled by running water and condensed. From this point it may be carried in uncovered pipes to any distance, as it is at ordinary atmospheric temperature. The expansion valve *E* is located near the brine tank or cold storage room, and here the liquid evaporates at the release

of pressure, taking from the surrounding medium the latent heat of evaporation and expansion. At atmospheric pressure the latent heat of evaporation for ammonia is 573 B.t.u.

The standard unit of refrigeration as generally used is equivalent to the melting of 2000 lb. of ice at 32 deg. Fahr. to water at the same temperature, or $142 \times 2000 = 284{,}000$ B.t.u. On account of various losses the amount of energy actually used corresponds to about 420,000 B.t.u. per ton of ice actually made.

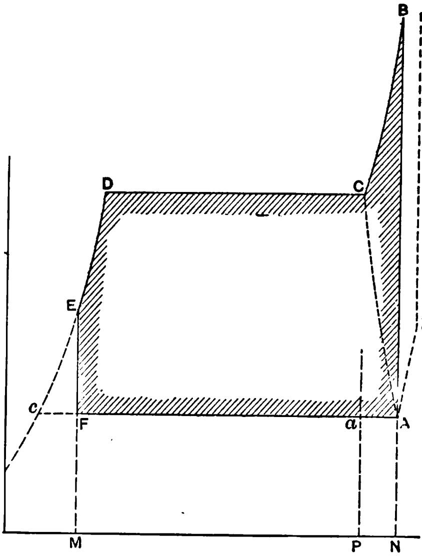


Fig. 34. Entropy Diagram for Refrigerating Cycle.

The entropy diagram is the one most suitable for studying the refrigerating machine, since the process is primarily a thermodynamic one. As might be expected, such a diagram, Fig. 34, is practically the same as that for a steam engine, but with a reversed cycle. Starting with the vapor at the lower temperature as it enters the compressor at *A*, we have compression which is more or less adiabatic to *B*, superheating the vapor. If the vapor is wet when it reaches the compressors then compression may be along some line like a *C* with no superheat.

On the other hand the gas may become superheated and reach the compressor at some such condition as *b*. *AC* is the line of saturation for ammonia; distances to the right of this show superheat and distances to the left condensation. In a water-jacketed compressor the compression would not be adiabatic, but along some line approximating to *AC* and heat would be given up to the jacket water.

The specific heat of ammonia gas at constant pressure is 0.52.

BC is the line of cooling to the point of saturation, *CD* the condensation at the higher temperature, and *DE* the cooling of the liquid in the pipes.

At *E* the expansion valve is opened, *EF* is adiabatic expansion, accompanied by evaporation *cF,* and *FA* the evaporation at the lower temperature. If the expansion at *E* is free expansion with constant heat the line *EF* will incline slightly to the right and some heat will be used in reducing the liquid to boiling point. As has been shown the range of temperature will be ordinarily between 80 deg. and —15 deg. Fahr. not including the peak at *B*.

The cycle is thus seen to be entirely below that of steam on the temperature scale.

The efficiency of the steam cycle is measured by the ratio of the shaded area to the entire area under the upper line. The object of improvements is to make the shaded area as large as possible, which is done by increasing T_1 and diminishing T_2.

In the refrigerating machine, on the contrary, the efficiency is measured by the ratio of the area under *FA* to the shaded area, since the former represents the heat extracted from the substance to be cooled and the latter the work expended. As Professor Reeve points out, this can hardly be called an efficiency, because these two areas are in a measure independent. The object, however, of improvements in this class of machinery would be to reduce the shaded area and the difference T_1—T_2 should be kept as small as practicable.

46. Actual Performance of Refrigerating Machines.—As has been already explained, the standard ton of refrigeration is 284,000 B.t.u. and the actual making of one ton of ice usually takes from 410,000 to 420,000 B.t.u.

In a paper read before the American Society of Mechanical Engineers, Mr. J. C. Bertsch makes the following recommendations for standard conditions as representing the averages obtaining in practice:

1. 284,000 B.t.u. as the latent heat of 2000 lb. of ice constitute one ton of refrigeration.

2. The efficiency of the ammonia compressor is 75 per cent of the theoretical capacity.

3. The limit of piston travel in feet per minute shall be:

180 ft. for strokes up to and including 12 in.

240 ft. for strokes over 12 and including 24 in.

300 ft. for strokes over 24 and including 36 in.

360 ft. for strokes over 36 in.

4. The temperature to be produced is 15 deg. Fahr. and the boiling point of the evaporating ammonia is zero.

5. The temperature of the condensing water is taken at 75 deg. Fahr. and the temperature of the liquid ammonia at 80 deg. Fahr.

6. The displacement of the compressor must be 5 cu. ft. or 8640 cu. in., per ton, per minute.

The calculation can then be made in this way:

Actual capacity of compression =

$$5 \times 0.75 = 3.75 \text{ cu. ft.}$$

$$= 3.75 \times 0.1094 = 0.41 \text{ lb. at 0 deg. Fahr.}$$

$$555.5 - (80 \times 1.23) = 457 \text{ B.t.u. per lb.}$$

since specific heat of liquid = 1.23

$0.41 \times 457 \times 24 \times 60 = 270{,}000$ B.t.u. per day or practically one ton of refrigeration in 24 hours.

But in addition to the actual freezing of the water at 32 deg. there is the cooling of the water from the temperature at which it enters the cans and the subsequent cooling of the ice to 15 deg. Assuming the water to be at a temperature of 80 deg. Fahr. the heat per pound will be

$$(80 - 32) + \tfrac{1}{2}(32 - 15) = 56.5 \text{ B.t.u.}$$

$$2000 \times 56.5 = 113{,}000 \text{ B.t.u. per ton.}$$

Adding this to the latent heat we have:

$$284{,}000 + 113{,}000 = 397{,}000 \text{ B.t.u.}$$

External losses will bring the total up to an average of 415,000

B.t.u.. The actual ice-making capacity of the machine just calculated would then be about:

$$\frac{270,000}{415,000} = 0.65 \text{ ton in twenty-four hours.}$$

In the wet system of compression some liquid ammonia is present in the compressor and the line of compression may follow aC or some line to the left of this. Superheat and its attendant high temperature is thus avoided and the work of compression is reduced. Experience, however, shows very little difference in efficiency between this and the dry method.

The horse power of the compressor under the conditions mentioned above would be approximately as follows: Assume suction at 15 lb. gage and upper limit of pressure as 140 lb. gage, corresponding approximately to the limits of temperature already given.

Assume the compression as adiabatic and use the equation $pv^n =$ a constant. Then n will be the ratio of $\frac{C_p}{C_v}$ and this for ammonia is

$$\frac{0.52}{0.41} = 1.27$$

$$\text{Ratio of compression} = \left(\frac{155}{30}\right)^{\frac{1}{n}} = 3.64$$

Initial volume $= 5$ cu. ft.
Final volume $= 1.37$ cu. ft.
Work of compression in foot pounds is by Art. 12.

$$A = \frac{np_2v_2 - p_1v_1}{n-1}$$

$$= \frac{144(1.27 \times 155 \times 1.37) - 144(30 \times 5)}{0.27}$$

$= 64000$ ft. lb. per minute.
$=$ approximately 2 h.p.

Problems.

1. Calculate the entropies of water and of steam at pressures of 7, 19 and 65 lb.

2. Steam at a pressure of 115 lb. expands on the saturated steam line to a pressure of 17 lb. and is condensed at the latter pressure. Draw the pv diagram for 1 lb. of steam, using a scale of 40 lb. to 1 in.

3. Draw the $T\phi$ diagram for Problem 2, using a scale of 50 deg. to 1 in. and one half a unit of entropy to 1 in. Omit the part of the figure below 212 deg. Fahr.

4. Draw the adiabatic lines for Problems 2 and 3 and determine the area of the Rankine cycle in the $T\phi$ diagram, expressed in heat units.

5. Assume the steam in the preceding problems to have 20 per cent of moisture during expansion and to expand only four times, draw the pv and $T\phi$ diagrams and determine the efficiency as compared with the Rankine cycle.

6. Assuming the same conditions as before but no expansion, draw the constant volume line and determine the efficiency. Calculate the heat received and rejected in different stages of the process in Problem 5.

7. Assuming the use of a condenser and air pump in Problem 5, so that the steam is condensed at a pressure of 3 lb., find the gain in heat units and in per cent.

CHAPTER V.

VALVE AND LINK MOTIONS.

47. Cylinder and Valve.—As has already been explained in Chapter II. the steam in an engine cylinder expands and does work by moving the piston, while the admission and release are controlled by the steam and exhaust valves.

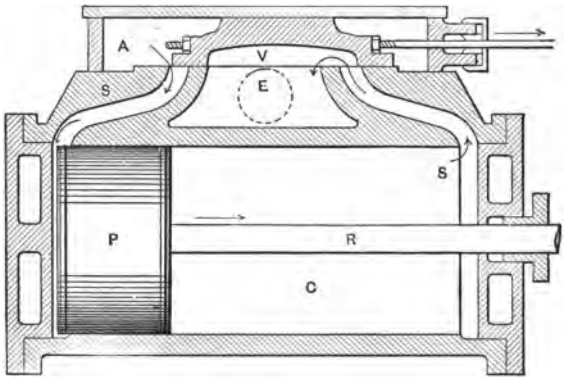


Fig. 35. Engine Cylinder and Slide Valve.

Fig. 35 illustrates the common arrangement of a cylinder with one slide valve controlling the admission and release of the steam at both ends of the cylinder. *C* is the cylinder proper with its piston *P* and piston rod *R*. *SS* are the steam ports connecting the interior of the cylinder with the steam chest *A*, while the exhaust port *E* communicates with the outside air or the condenser.

The slide valve *V* moves longitudinally inside the steam chest, admitting steam alternately to either end of the cylinder. Steam is admitted to the chest by means of the steam pipe and throttle valve. (See Fig. 7.) The valve shown in Fig 35 is known as a D-valve and controls both the entrance and exit of the steam.

For instance, when in the position shown in the figure, the valve admits steam to the left end of the cylinder and at the same time allows steam to escape from the right end through the valve cavity and the exhaust port *E*.

The piston and valve are now moving to the right as shown by the arrows and the left steam port is opened more and more as the

piston advances. When the piston has moved a short distance the valve returns, closing the left port and later preventing the escape of steam from the right port. When the piston has reached the right end of its stroke, the valve will have moved to the left far enough so that the right steam port will be slightly

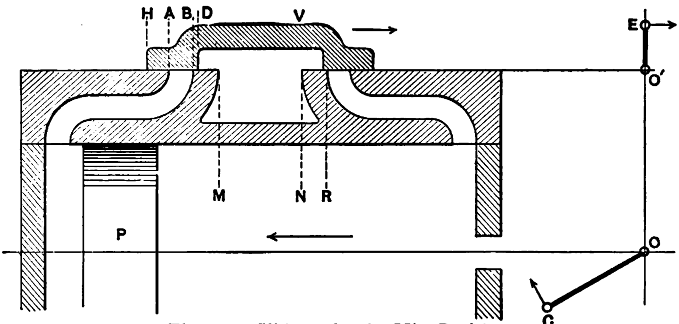


Fig. 36. Slide Valve in Mid Position.

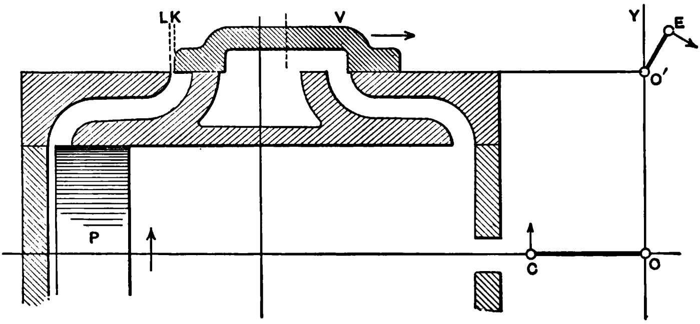


Fig. 37. Admission of Steam.

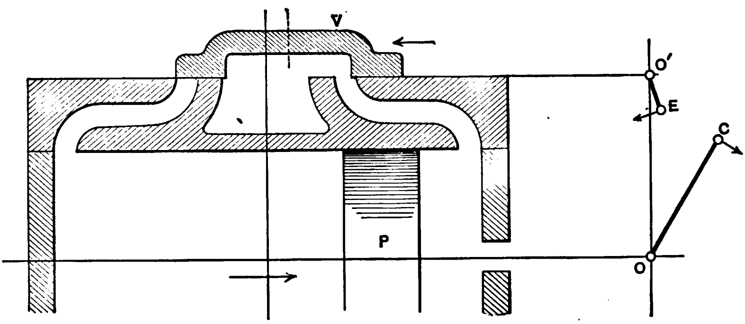


Fig. 38. Cut-off.

open to steam and the exhaust steam will be escaping at the left end.

It is evident that the behavior of the steam will depend upon two things, the proportions of the valve in relation to the ports and the location of the crank or eccentric which moves it relatively to the main crank.

48. Motion of the Valve.—Figs. 36, 37 and 38 show different positions of the piston and valve and the crank which connects them. In each figure OC is the center line of the main crank which is driven by the piston P, while $O'E$ is the radius of the eccentric which usually takes the place of a crank for driving the valve V.

The centers O and O' coincide in the actual engine, but in the figure they are located for convenience on the lines of stroke of the piston and valve respectively.

Before tracing the relative motions of piston and valve it will be necessary to study the proportions of the latter.

Fig. 36 shows the valve in middle position, its center line coinciding with that of the steam chest, and its edges lapping over both ports. When in this position the distance AH which it overlaps the port on the steam side is called the steam lap and the corresponding overlap BD on the exhaust side is called the exhaust lap. These are sometimes called outside and inside laps, but such terms are indefinite, since many valves take steam on the inside and exhaust on the outside.

The following terms and symbols will be used hereafter in describing valve motions:

Crank radius $= OC = R = \dfrac{L}{2}$
Eccentric radius $= O'E = r$
Width of steam port $= AB = w$
Width of exhaust port $= MN = W$
Width of bridge $= NR = b$
Steam lap $= AH = l$
Exhaust lap $= BD = e$
Steam lead $= LK = d$ (See Fig. 37.)
Maximum port opening $= m$
Maximum exhaust opening $= n$

The term *travel* will be used arbitrarily to denote the distance of the valve from its middle position at any time, and will equal x. In Fig. 36, since the valve is central, the eccentric radius will be vertical. The various parts will be moving as shown by the arrows, and the valve is soon to admit steam on the left. In Fig. 37, the piston has reached the left end of its stroke and the valve is open an amount KL which is called the *lead*. The crank OC is now horizontal and $O'E$ has passed the vertical by an angle $YO'E$, which is called the angular advance and which we will denote by α.

The angle between the crank and eccentric radii therefore = 90 deg. + α measured from the crank in the direction of rotation. This angle depends upon the amount of steam lap and increases with it.

The travel of the valve evidently = steam lap + lead or $x = l + d$.

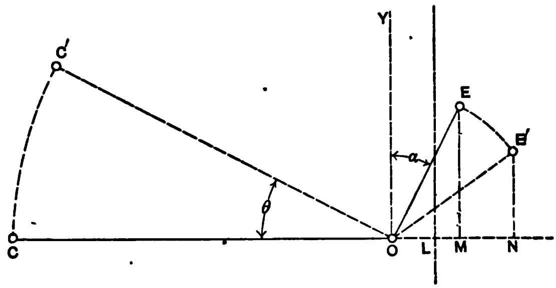


Fig. 39.

As the crank turns the piston now moves to the right, the valve opens wider, reaches the limit of its travel and returns to the position of *cut-off* shown in Fig. 38. In this position $x = l$.

The functions of the exhaust side of the valve are as follows: When the valve has traveled an amount e to the right in Fig. 36, the right exhaust port is opened and *release* occurs. The exhaust port remains open while the valve goes to the limit of its travel and returns again to the same position $x = e$, when the port is closed and *compression* begins on the right side of the piston, forming a cushion to absorb the momentum of the moving parts.

While the crank turns another half revolution the piston moves from right to left and the right end of each valve face comes into action.

49. Valve Diagrams.—In Fig. 39 let OC and OE be positions of crank and eccentric when the former is on a dead point, and let OC' and OE' be their positions after the crank has turned through an angle $= \theta$. Then will the travel of the valve $= x = ON$. Next draw a vertical line through L so that $OL = l =$ the steam lap. As it is necessary for the valve to travel a distance $= l$ in order to open the steam port, the width of port opening at any time $= x - l$. When the crank is at C' the port opening $= ON - OL = LN$ and when the crank is at C, $OM - OL = LM =$ the lead $= d$. (See Fig. 37.) The position of the valve can thus be found for any position of the crank, but the process is a tedious one.

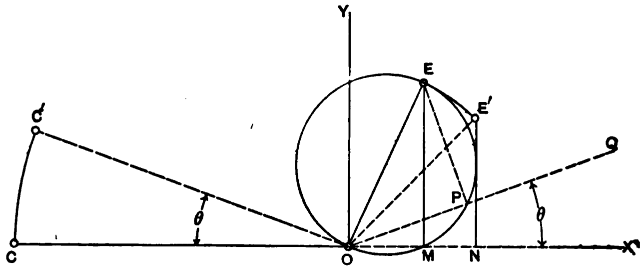


Fig. 40.

An easier and better way is to use some one of the numerous forms of valve diagrams which have been invented for this purpose. The one best known is the Zeuner diagram and this will be used here since it is the best known and as convenient as any. In Fig. 40 OC and OE are the initial positions of crank and eccentric, while OC' and OE' are the positions after the crank has turned through an angle θ. The corresponding travel of the valve is $ON = x$. Now describe a circle on OE as a diameter and draw a line OQ, making the angle $QOX = \theta$. OQ cuts the circle at P and the triangles OPE and ONE' are similar and equal. Hence $OP = ON = x$.

To find the travel of the valve for any crank position we will then proceed as follows: Describe a circle, called the valve circle,

on the initial position of the eccentric radius as a diameter. Turn the axis opposite the crank backwards through angles equal to the successive crank angles; then will the intercepts cut by the valve circle from the axis represent the corresponding valve travels.

In Fig. 41, let OE be the radius of eccentric and YOE equal angular advance. Let two valve circles be drawn in opposite quadrants on OE and OE'. With radii equal to l and e respectively draw the arcs ADB and HLF about O, cutting the two

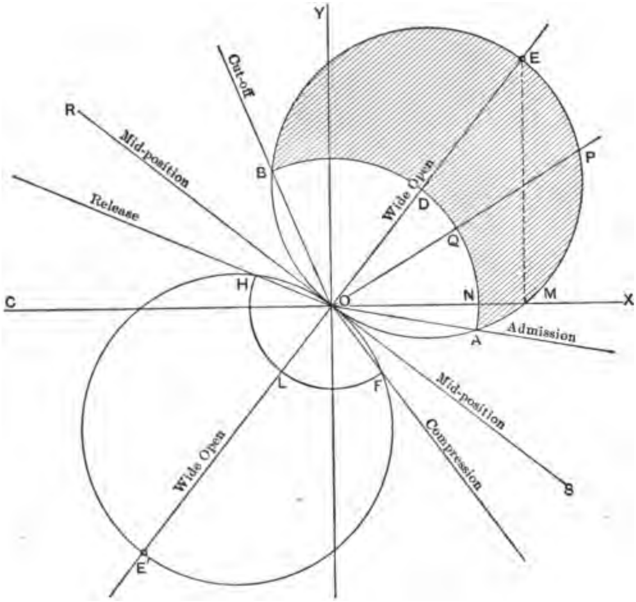


Fig. 41. Zeuner Valve Diagram.

valve circles as shown. ADB and HLF are known as lap circles. The crank OC is regarded as turning clockwise and the axis OX is therefore revolved contra clockwise.

Beginning with the position OA, the travel of the valve is equal to the steam lap and is increasing, therefore this is the beginning of admission. At the dead point OX the travel is OM and the port is open an amount $MN =$ the lead. At any position, as OP, the port is open an amount QP, called the port opening. The

principal positions of the valve are indicated in the following table, which may be readily interpreted in the light of what precedes:

Analysis of Valve Motion. (See Fig. 41.)

Angle turned by crank.	Travel of Valve.	Port Opening	Function.	Remarks.
$-XOA$	$OA=l$	None.	Admission.	XOA=lead angle.
None.	$OM=l+d$	$NM=d$	Lead.	Dead point.
XOE	$OE=l+m$	$DE=m$		Extreme travel.
XOB	$OB=l$	None.	Cut-off.	Diminishing travel.
XOR	None.	None.		Mid position.
XOH	$-OH=e$	Exh. opens.	Release.	Valve to left of center.
COE'	$-OE'=e+n$	$LE'=n$		Extreme travel.
COF	$-OF=e$	None.	Compression.	Diminishing travel.
COS	None.	None.		Mid position.

It is more convenient, though not necessary, to use the upper circle for steam and the lower one for exhaust functions. As used in the figure the circles give all the valve functions for one

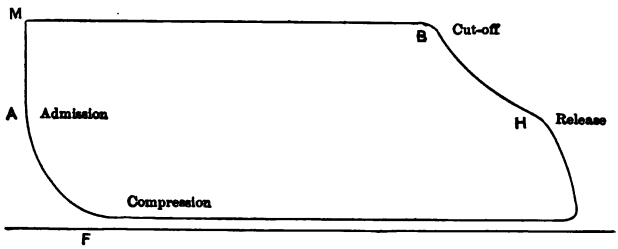


Fig. 42. Indicator Diagram Corresponding to Fig. 41.

end of the cylinder. The indicator diagram in Fig. 42 shows what occurs in the cylinder at the same time, the letters corresponding to those in Fig. 41. The maximum port opening, m, in the table may be greater or less than the port width, w, at the judgment of the designer.

50. Crank and Piston Positions.—So far we have considered the motion of the valves as referred to the rotation of the crank alone, without locating the positions of the piston.

In designing or in setting a valve it is usually necessary to refer it to the motion of the piston. If the errors in the crank motion

due to the obliquity of the connecting rod be neglected, the piston positions may be found by dropping perpendiculars from the crank-pin centers on the line of stroke as C_1M, C_2N in Fig. 43. That is, when the crank has turned through an angle COC_1 the piston will have moved a distance CM from the starting point. This method is quite generally used in designing valves, as it gives correct average values. If it is desired, however, to make allowance for the action of the connecting rod, this may be readily done by substituting for the perpendiculars C_1M, etc., circular arcs C_1P, etc., having radii equal to the length of the rod $FC_1 = AC$. The true position of the piston will then be F or H, so that $AF = CP$ and $BH = DQ$.

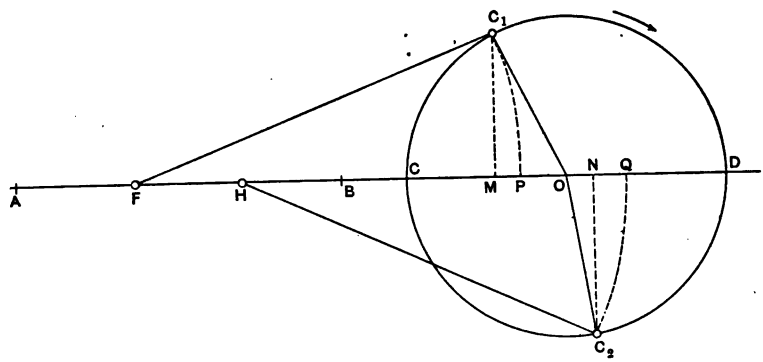


Fig. 43. Effect of Angularity of Connecting Rod.

Conversely, if the position of the piston F be known, the corresponding position of the crank C_1 may be found by drawing an arc like PC_1. If the length of the connecting rod is inconveniently great a template may be made to fit the arc and used on all crank positions.

51. Design of a Slide Valve.—The data commonly assumed in designing a valve motion are the port width and times of admission, cut-off and compression. For example, let us assume:

Port width $= w = 0.75$ in.
Angle of lead $= 10$ deg.
Average cut-off at 0.75 stroke.
Average compression at 0.9 stroke.

It is required to determine the radius and angular advance of the eccentric and the lead and laps of the valve. Draw a circle with a radius OC (Fig. 44) to represent the crank circle at any convenient scale. On the horizontal diameter lay off

$$XQ = 0.75XC$$

and

$$CR = 0.9CX.$$

Erect the perpendiculars QP and RK and draw the crank positions OP and OK for cut-off and compression. Lay off the lead

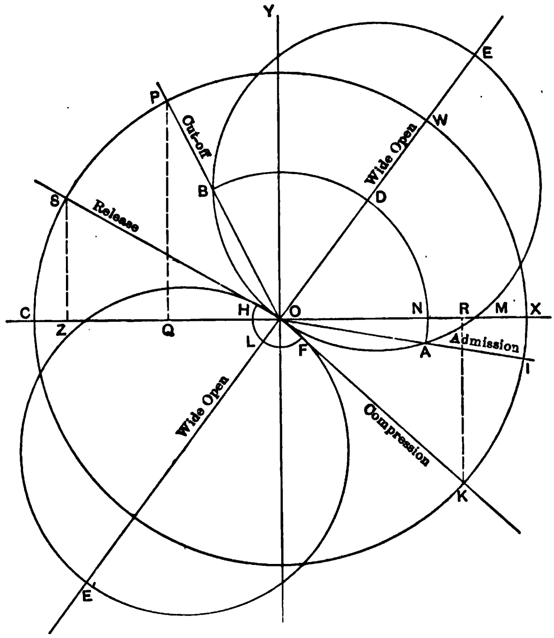


Fig. 44. Diagram Applied to Design of Valve.

angle $XOI = 10$ deg., then will OI be the crank position for admission. Bisect the arc IWP and draw the indefinite line $E'OWE$. This line will be the crank position for the mid-position of the valve and therefore will coincide with the diameters of the two valve circles. (See Art. 49.) Draw one trial valve circle

having a diameter less than twice an assumed maximum port opening. This latter is usually equal to or greater than the width of port. We will assume $m = 0.8$ in. Draw a trial lap circle through the intersection of the trial valve circle with OI or OP and find trial port opening $= r' - l'$. We can now find the true value of the eccentric radius by proportion:

$$\frac{\text{Trial port opening}}{\text{Trial radius}} = \frac{\text{True port opening } (m)}{\text{True radius } (r)}$$

Draw the two valve circles with diameters $OE' = OE = r$ intersecting the various crank positions as shown in the figure. As already explained in Art. 50, we have the following determinations:

$$\begin{aligned} Y'OE &= \text{angular advance} = 29 \text{ deg.} \\ OE &= \text{eccentric radius} = 1.4 \text{ in.} \\ OA &= \text{steam lap} = 0.6 \text{ in.} \\ OF &= \text{exhaust lap} = 0.05 \text{ in.} \\ NM &= \text{steam lead} = 0.2 \text{ in.} \\ DE &= \text{max. port opening} = 0.8 \text{ in.} \end{aligned}$$

Through F describe the circular arc FLH and draw the line OHS. Then $OF = e$, and OS will be the crank position for release; and if we drop the perpendicular SZ, $\frac{XZ}{XC} =$ fraction of stroke completed at release.

In order to draw the valve it will be necessary to assume the bridge width b and the width of exhaust port W (Fig. 45). The width b is equal to or greater than the thickness of the metal of the cylinder walls. W must be large enough so that when the valve is at the limit of its travel as shown by the dotted lines, the distance y may be at least equal to the port width w, in order that the flow of the exhaust may not be cramped.

The valve is drawn as follows: Locate the center line YY. Lay off on either side the distances $W/2$, b, w and l; their sum will be the total width of valve. Lay off the exhaust lap, e, and draw the exhaust cavity. The width of each valve face $= l + w + e$ and the width of the cavity $= W + 2b - 2e$.

52. Adjustments for Equal Cut-offs.—A valve designed as in the preceding article will cut off steam at equal crank angles in the two strokes, but the piston positions will not be alike.

The end of the cylinder next to the crank is called the *crank end* and the other the *head end.* As the piston leaves the head end and approaches the crank it is said to be making its *forward stroke,* while the reverse motion is called the *return stroke.* A crank which throws over (see Art. 17) is above the center line during the forward stroke and below during the return.

Reference to Fig. 43 will show that during the forward stroke the piston is ahead of its mean position, as at P, while during the return it is behind, as at Q. Consequently a valve designed as in Figs. 44 and 45 will cut off too late on the forward stroke and

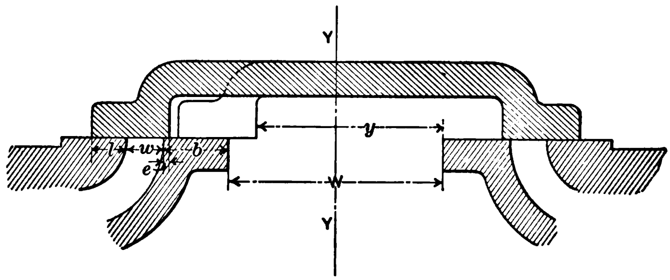


Fig. 45. Valve Designed by Aid of Fig. 44.

too early on the return. This cannot be remedied by moving the eccentric, since turning the eccentric ahead will make all of the functions of the valve occur earlier in both strokes and turning it back will have the reverse effect.

The head end of the valve does the cutting off during the forward stroke and the crank end during the return. The cut-offs can thus be equalized by adding a certain amount to the head steam lap and subtracting a similar amount from the crank lap. These amounts can be found by the Zeuner diagram as in Fig. 46, which is partly reproduced from Fig. 44. Q represents as before the position of the piston at cut-off. Draw two connecting-rod arcs QP_1 and QP_2 on opposite sides of the perpendicular QP and draw the two crank positions OP_1 and OP_2. XOP_1 is the crank angle at cut-off during the forward stroke and XOP_2 the cor-

responding angle for the return. OB_1 is then the steam lap for the head end of the valve and OB_2 the lap for the crank end, which give equal cut-offs on the two strokes. This equality has, however, been secured at the expense of equality of lead. Adding to the lap on the head end of the valve must delay the admission at that end and the reverse effect will occur at the crank end. This is shown in Fig. 46 by describing arcs with radii OB_1 and OB_2 and drawing new admission lines OI_1 and OI_2.

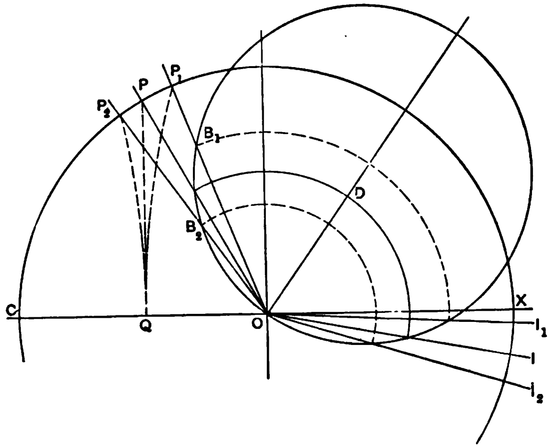


Fig. 46. Determining Adjustment for Equal Cut-off.

The lead is seen to be nearly zero on the forward stroke, while at the beginning of the return stroke it is almost double its former value. Inequality of lead is usually considered by engineers a more serious disadvantage than inequality of cut-off, and valves are usually designed as in Fig. 44.

53. Lengthening the Valve Stem.—A compromise is sometimes made by designing a valve which is symmetrical, as in Fig. 45, and then lengthening the valve stem so that the lead will be less at the head end and the cut-off will be partially equalized.

This adjustment affects the action of the exhaust edges of the valve in a similar manner, tending to equalize the compression and to disturb the release.

54. Setting Eccentric by Zeuner Diagram.—If the dimensions of the eccentric, valve and ports are given, all that can be

done is to so set the eccentric as to secure either a certain lead or a certain cut-off.

In Fig. 47 describe the steam lap circle with a radius $ON = l$ and draw the axis OX and the crank position for cut-off OP. Then OB must be a chord of the valve circle. Since the radius of the circle is known the center K may be found and $KOY = \alpha$, the angular advance.

If a certain lead d is desired, lay off $NM = d$, use OM as a chord and find the center of the valve circle as before.

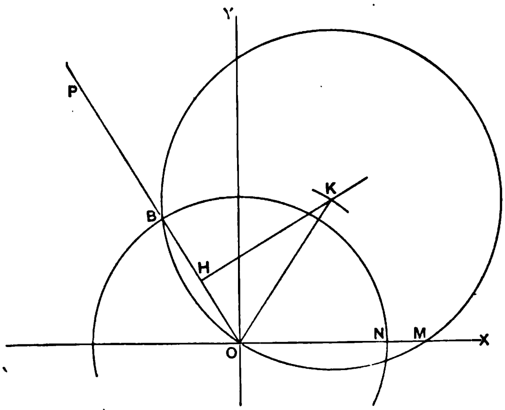


Fig. 47. Locating the Eccentric.

55. Location of Eccentric.—The Zeuner diagram shows the correct angle between the crank and eccentric, but it does not show in which direction to lay off that angle. This must be determined by the mechanism and the direction of motion of the particular engine under discussion.

In Fig. 48 let OC be the crank on a dead center and OE the corresponding position of the eccentric. If the connection between the eccentric and valve is a direct one, positions 1 and 2 are correct for the two directions of rotation indicated by the arrows. If the motion of the valve is reversed by the interposition of a rock-arm, as is common in locomotives, positions 3 and 4 are correct.

It is thus seen that the eccentric may be located in any one of the four quadrants according to circumstances. The same Zeuner diagram could be used for any one of the four cases

shown in Fig. 48. It is important to remember this fact and to understand that it makes no difference whether we call OC or OX the crank in the diagrams such as are illustrated in Figs. 41, 44, etc.

A valve which admits steam on the inside and exhausts at the ends has a motion the reverse of the ordinary D-valve, and if driven directly would have eccentric positions like 3 and 4 in Fig. 48. Piston valves are usually made in this way, having the steam lap inside the valve and the exhaust lap at the end.

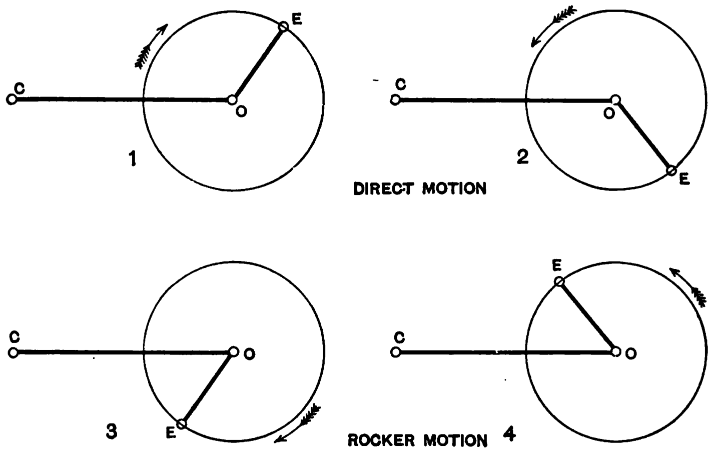


Fig. 48. Location of Eccentric under Different Conditions.

Some valves have no lap and are moved by an eccentric at right angles to the crank. These are sometimes called square valves and the indicator diagrams from an engine thus equipped will be nearly a rectangle.

56. Shifting Eccentric.—It is sometimes desirable to shift the position of an eccentric on the shaft in order to change the point of cut-off or even to reverse the engine. In doing this it is better to keep the lead nearly constant, so that the time of admission will not be affected seriously.

In other words the center of the eccentric should be moved nearly on the line EM in Fig. 41. Figs. 49 and 50 illustrate two

methods of effecting this. In Fig. 49, *E* is the slotted eccentric while *F* is a flange keyed to the shaft. The eccentric is fastened in position by the nut *N*, and it is necessary to stop the engine in order to make any adjustment. In Fig. 50 the eccentric disc *E* is

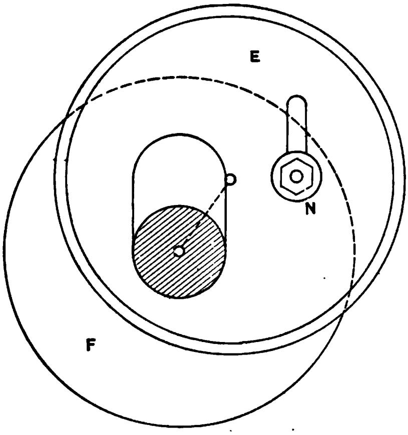


Fig. 49. Shifting Eccentric on Straight Line.

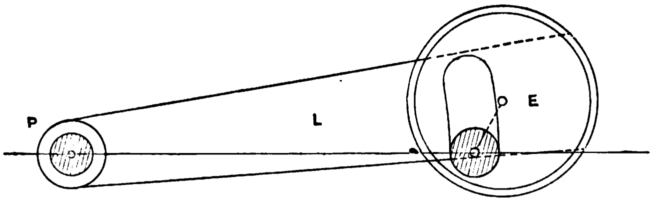


Fig. 50. Shifting Eccentric on Arc of Circle.

fastened to the face of a lever *L* which can be turned about pin *P*. *P* is usually fastened to the arm or rim of the fly wheel and the lever *L* is controlled by a governor. (See Chapter VIII.) The eccentric is slotted for the shaft and moves in a rather flat circular arc. The lead will increase slightly as the center moves down.

Figs. 51 and 52 show the Zeuner diagrams for the two arrangements just explained. In Fig. 51 the center of the eccentric moves down from *E* to *E'*, the cut-off changes from *OP* to *OP'* and the

lead remains constant at NM. The travel of the valve is of course reduced to OE'. In Fig. 52 the center E moves on the dotted arc to E', shortening the travel the same amount as before.

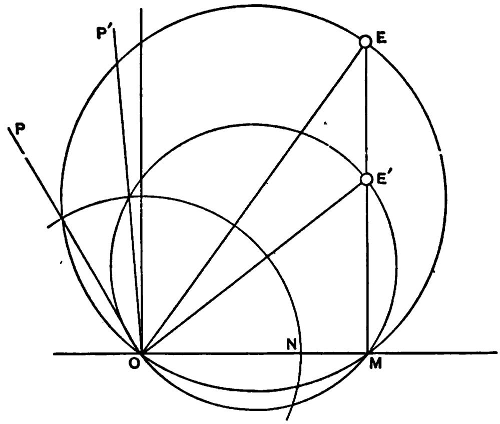


Fig. 51. Diagram for Fig. 49.

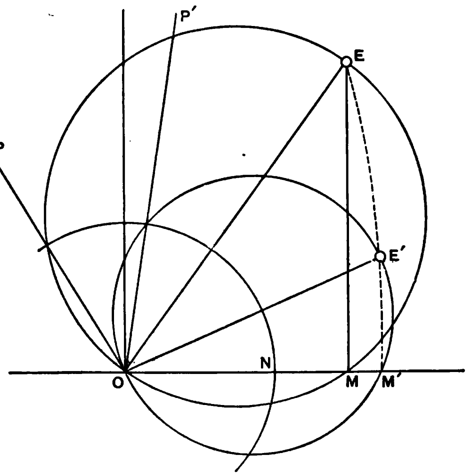


Fig. 52. Diagram for Fig. 50.

The angular advance is increased more than in Fig. 51 and the cut-off position OP' is earlier, while the lead is increased to NM'. By elevating the location of P in Fig. 50 a little above the center

line, the arc EM' may be made to approximate more closely to the vertical EM, and there will be less variation of lead.

57. The Stephenson Link.—It is evident that if the center of the eccentric in a shifting motion, like either of those just described, is carried below the horizontal line OX, the valve will move in the opposite direction and the tendency will be to reverse the engine.

There is difficulty in doing this successfully when an engine is in motion and various so-called *link motions* have been invented to

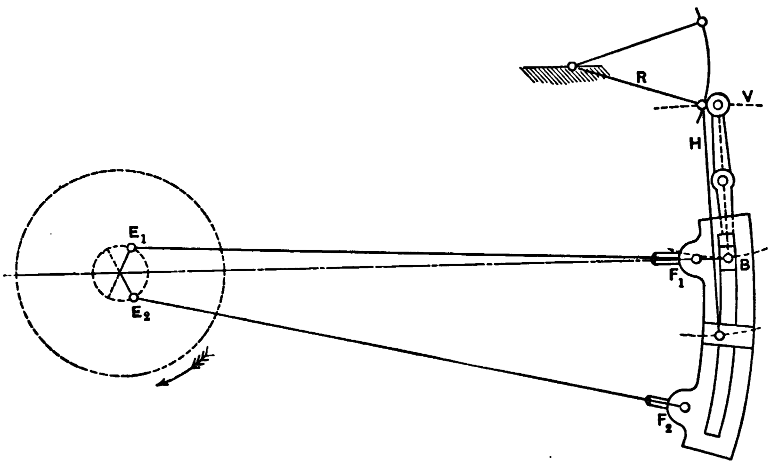


Fig. 53. Stephenson Link Motion.

serve the same purpose. The best known and most widely used is the Stephenson link, so called from the name of its inventor.

Fig. 53 shows the main features of this mechanism in skeleton. Two eccentrics, E_1 and E_2 are keyed to the shaft in such positions that one would drive the valve for forward motion of the engine and the other for backward motion, if either were used separately. For instance, in the figure, E_1 would serve for clockwise motion and E_2 for contrawise. The ends of the eccentric rods F_1 and F_2 are attached to the two ends of a curved link. The link is slotted and a block B slides freely in the slot; this block moves the valve stem V, either directly or through a rock-arm as shown.

The link is suspended by a hanger *H*, attached usually to some point near its center, and this hanger is raised or lowered by the rock-arm *R*.

When the link is in mid-position, the block *B* receives motion equally from both eccentrics such that the engine will not run in either direction. When the link is dropped to its lowest position, as in the figure, so that F_1 and *B* are on a level, the valve will be controlled almost entirely by E_1 and the engine will run clockwise. The motion can be reversed by raising the link to its highest position until F_2 and *B* are on a level and the valve is controlled by E_2.

In intermediate positions of the link the valve will receive a motion similar to that produced by a shifting eccentric. The motion of any point in the link is an aggregate one made up of the motions received from the two eccentrics and the motion of the point of suspension where the hanger is attached. The path can be proved by construction to be an elongated figure eight.

58. Slip of the Link.—The block *B* is usually guided by a rock-arm with the center above *B* and therefore moves in a circular arc, concave above. The center of the link moves in a similar path about the upper end of the hanger. When the link is in the middle position, it and the block move nearly together. When the link is in either of the extreme positions the motions of the block and link are so entirely different that there is considerable slipping between the two. This is a disadvantage, as it causes wear between the two and lost motion. Since the slip is least near the point of suspension, it is good design to locate that point near the part of the link which will be most used. The link motion is employed on marine engines mainly for reversing, and is either in full forward or full backward gear. The point of suspension is accordingly located near the forward end of the link, to reduce the slip at that point.

In locomotives the link motion is used for varying the cut-off and grade of expansion and the point of suspension is usually near the center of the link.

59. Open and Crossed Rods.—When the centers of the eccentrics are between the shaft and the link the rods may be open as in (1) or crossed as in (2), Fig. 54. These positions

determine the name of the motion, and that shown in (1) is called an open rod motion and that in (2) a crossed rod motion. The position of the crank makes no difference and depends only upon the kind of valve used and the type of connection, as explained in Art. 55.

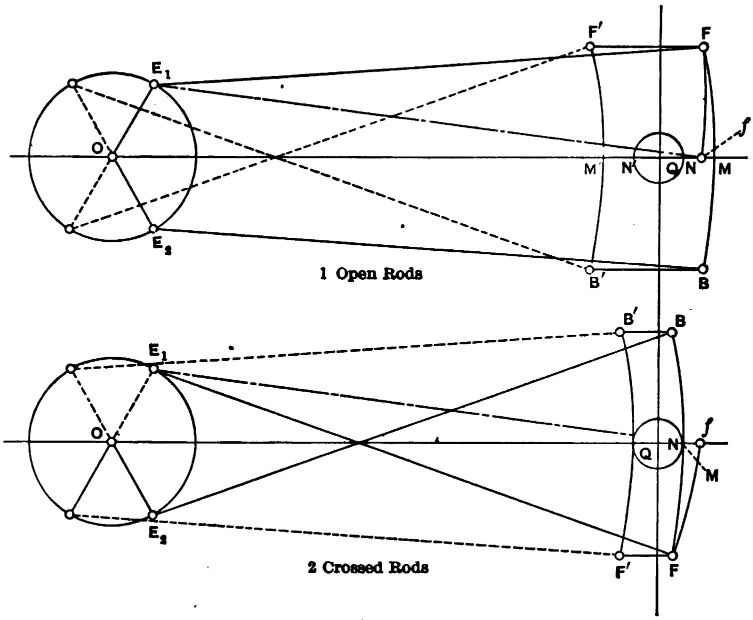


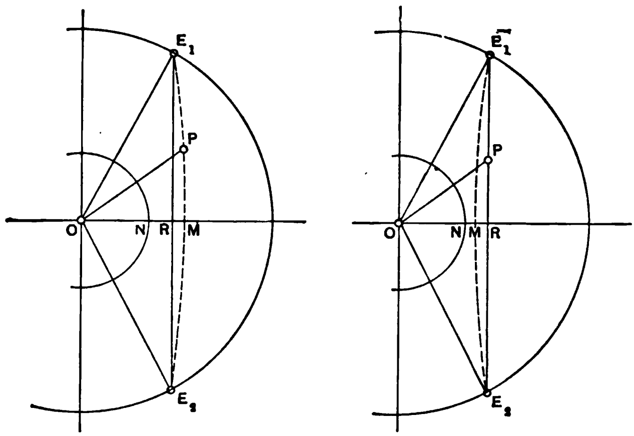
Fig. 54. Open and Crossed Rods.

The action of the valve when the link is shifted is affected in a marked degree by the difference in the arrangement of the rods. When the open rods are used the lead of the valve is greatest with the link in mid-gear; with the crossed rods the mid-gear lead is less than that at full gear. This may be easily proved by construction as in the figures. In both diagrams F and B are the forward and backward ends of the link and M the middle. FMB is the mid-gear position of the link for one dead point, and $F'M'B'$ the mid-gear position for the opposite dead point. Bisect $M'M$ at Q, then QM is the travel of the valve when the link is at mid-gear. This distance is greater in (1) than in (2). On Q as a

center describe a circle with a radius $QN =$ the steam lap. NM is the mid-gear lead and is practically equal to 0 in (2).

About E_1 as a center swing the eccentric rod E_1F to the center line at f, then Nf is the lead for full gear forward and is the same in both figures. In (1) the lead increases the amount fM as the link changes from full to mid-gear, while in (2) the lead decreases a similar amount. The curvature of the link is such as to equalize the leads on the right and left of the center. If the curvature of the link in Fig. 54 were increased, the points M and M' would both be moved to the right, the lead NM would be increased and the lead $N'M'$ diminished.

The radius of curvature can be readily found by construction and is approximately equal to the eccentric rod E_1F.



(1) Open Rods (2) Crossed Rods

Fig. 55. Zeuner Diagram for Link Motion.

60. Zeuner Diagram for Link Motion.—The Stephenson link is best designed by construction on a large scale, using template or models. The radius of the link, the location of the point of suspension and the position of the hanger all have an influence on the motion of the valve block and need to be determined accurately. The Zeuner diagram can only be used to give an ap-

proximation to the motion of the valve in making a preliminary study.

Fig. 55 shows the eccentric centers E_1 and E_2 similar to those in the preceding figure, (1) for open rods, (2) for crossed rods. If the rods had no obliquity the motion of the middle of the link would be the same as if produced by one eccentric having a radius OR and the lead would be constant and equal to NR. The effect on the valve motion of sliding the link up and down would be the same as that produced by shifting the eccentric in Fig. 51.

But it has been shown in Fig. 54 that the lead is changed by raising or lowering the link. Determine in (1) and (2) respectively (Fig. 54) the mid-gear travel QM and lay it off to scale on (1) and (2) in Fig. 55 as OM. Through E_1ME_2 pass a circular arc and it will be the locus of the center of a shifting eccentric which would give approximately the same motion to the valve as do the combined eccentrics and link.

Valve diagrams drawn on the respective radii as in Figs. 51 and 52 will show the nature of the valve motion. It can be thus shown that the open rods give larger port openings for a given range of cut-offs, and that this arrangement is better adapted to engines where a variable cut-off is desired. The crossed rods are well suited to engines which are to be stopped or reversed by the link motion as in hoisting machinery.

61. The Walschaert Valve Gear.—For many years the Stephenson link motion has been the standard gear in American locomotive practice, but recently some of the leading railroads of the country have been equipping their engines with the Walschaert gear, and there is a prospect that it may come into general use. We find it mentioned in Auchincloss' standard work on valve gears as far back as 1869, and of late the appearance of Mr. Wood's treatise devoted to this subject has brought it down to date. Fig. 56 shows the general arrangement of the Walschaert gear and is taken in the main from Mr. Wood's book.

The valve stem V is driven by a combination lever C, the lower end of which is fulcrumed to the crosshead of the engine through the medium of a short link S. Near the upper end of the lever C, a radius rod R is attached and this in turn is driven by a

link *L*, curved to the radius of the rod. The link is fulcrumed at its center *F* and is actuated at or near its lower end by an eccentric rod *E*. The eccentric rod is driven by an eccentric or crank set 90 deg. ahead of the main crank. By means of the usual reverse lever, the radius rod can be raised or lowered so as to vary the cut-off or reverse the engine.

The combination lever in this mechanism has evidently an aggregate motion produced by the two drivers *R* and *S*. The motion of the valve stem due to *S* is similar to that of the crosshead on a reduced scale and is the same as would be produced by an eccentric having an angular advance of 90 deg. (i.e. opposite the crank).

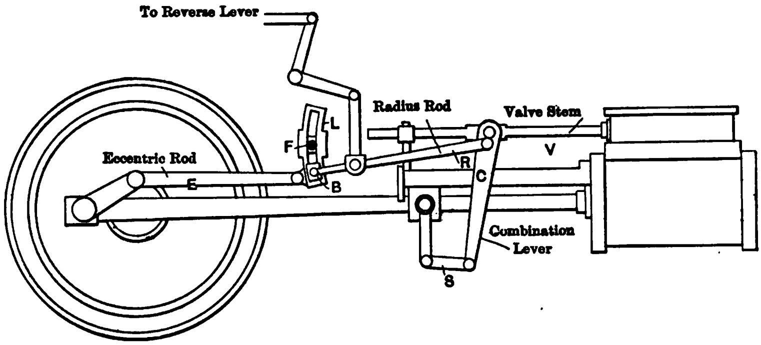


Fig. 56. Walschaert Valve Gear.

The motion due to *R* is almost directly that of the eccentric 90 deg. ahead of the crank, modified as to amount by the position of the block *B* in the link.

When the crank is on the left dead center, as shown in the figure, the valve should be to the right of the middle position an amount equal to lap + lead; as the lower end of the combination lever is now at full throw, the segments of the lever should have the ratio

$$\frac{R}{lap + lead}$$

(where R = crank radius).

The radius of the eccentric is responsible for the further movement of the valve and should be sufficient to give the port opening

desired when the link block is in full forward position. This amount is more readily determined by the aid of the Zeuner diagram (Fig. 57).

Let $ON =$ the steam lap of valve.

$NM =$ the desired lead.

Then will OM be the motion to be obtained by the connection to the main crosshead and this distance will determine the segments of the combination lever. Erect the perpendicular ME and with a radius $= ON +$ desired maximum port opening intersect ME

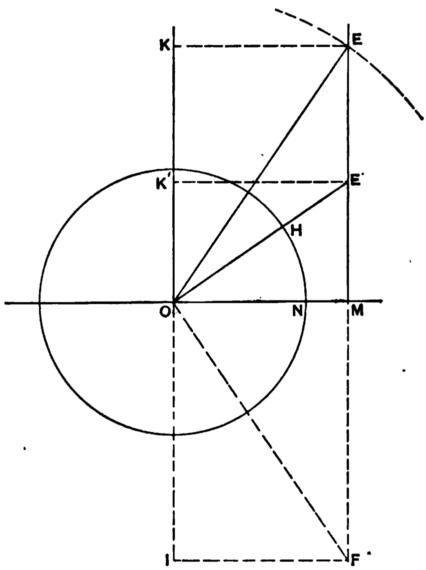


Fig. 57. Diagram for Walschaert Gear.

at E. Then will OE represent the maximum travel of the valve from its middle position and also the position and extent of the radius of the effective or resultant eccentric. Draw EK parallel to OM. Then will OK represent the radius of the actual eccentric which is to drive the link and radius arm, as shown in Fig. 56. The lead of the valve is constant because if the link is properly curved and in middle position, as in the figure, the block can be moved from end to end of the link without affecting the position of the valve. The effect of thus moving the link block is merely

to reduce the motion produced by the eccentric OK to some value as OK' (Fig. 57). The corresponding resultant OE' gives the actual travel of the valve and HE' will now be the new port opening. Raising the link block to the upper end of the link will change OK to OI, giving a resultant eccentric OF and therefore reversing the motion of the engine. The action of the Walschaert gear is thus similar to that of a shifting eccentric which moves directly across the shaft in a straight line. There are some modifications which have to be made in the motion to obviate certain errors in the travel of the valve. Instead of attaching the eccentric rod to the lower end of the link, it is usually attached to a projection extending below the link. This brings the mean position of the eccentric rod more nearly parallel to the center line of the engine and the travel of the valve is less than it was before. In the combination of two harmonic motions differing 90 deg. in phase, this gear is similar to the Allen link so much used on Porter-Allen engines. (See Chapter VIII.)

Where it is not possible to bring the point of attachment of the eccentric rod to the link low enough for the equalization of motion, the same result is sometimes accomplished by increasing or diminishing the angular advance of the eccentric. In some locomotives, it is necessary to put the link at a considerable height above the center line of the engine and in such cases this latter method of equalizing is adopted. The point of suspension of the hanger which moves the radius rod has some influence on the motion of the valve and causes more or less slip of the block in the link. There are necessarily errors in this motion, as there are in the Stephenson motion, due to the use of oscillating levers, and it is necessary to correct these by trial and adjustment on drawings made to a large scale. The advantages of the Walschaert gear, as compared with the Stephenson, are summarized by Mr. Wood in his treatise, essentially as follows:

1. Accessibility for inspection, lubrication and repairs.
2. Weight. A saving of 35 or 40 per cent in the weight of the gear is often effected by substituting this motion for the Stephenson. (This is doubtful.)
3. Directness of action. The construction of the new gear gives more direct transmission of motion.

4. Permanence of adjustment, due to the use of a crank instead of large eccentrics.
5. Wear. The rubbing surfaces of this gear are more readily lubricated and kept in repair than those of the Stephenson link.
6. Smooth operation, due to less angularity of the links.
7. Frame bracing. The removal of the valve gear from between the driving wheels permits bracing of the frame of a locomotive.
8. Constant lead.

For a complete discussion of this type of gear with methods of design, etc., the reader is referred to Mr. Wood's treatise in this subject.*

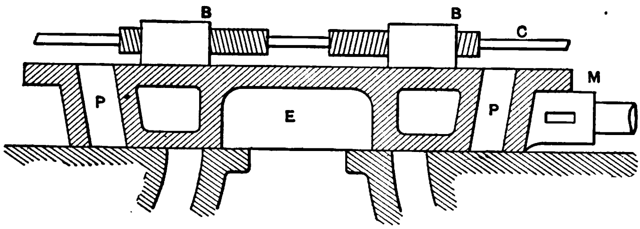


Fig. 58. Meyer Valve.

62. The Meyer Riding Cut-off Valve.—In marine engines the main slide valve controlling the admission and release of the steam is frequently driven by a crossed rod link motion so that the engine may be quickly reversed, but the variation of cut-off is effected by a second valve riding on the back of the main valve and driven by a separate eccentric.

Fig. 58 shows the principle of this arrangement. *M* is the main valve and is in effect an ordinary slide valve, having the usual exhaust cavity and two faces with exhaust and steam laps. This valve is designed in the usual way to give proper lead and compression and to cut off rather late, say at ¾ or ⅞ stroke. Two steam ports *P P* are then formed at the outer end of the valve, of

* "The Walschaert Locomotive Valve Gear," by W. W. Wood.

about the same width as the ports in the valve seat. The top of the valve is planed and two cut-off blocks *B B* driven by one valve stem *C* are arranged to slide on the planed surface and control the admission of steam to the ports *P P*. Right and left threads are cut on the upper valve stem, so that by rotating the stem the distance between the two blocks can be changed.

The dimensions of the blocks and the positions of the two eccentrics can be most easily determined by the aid of the Zeuner diagram. In Fig. 59 let *OE* be the radius of the main eccentric

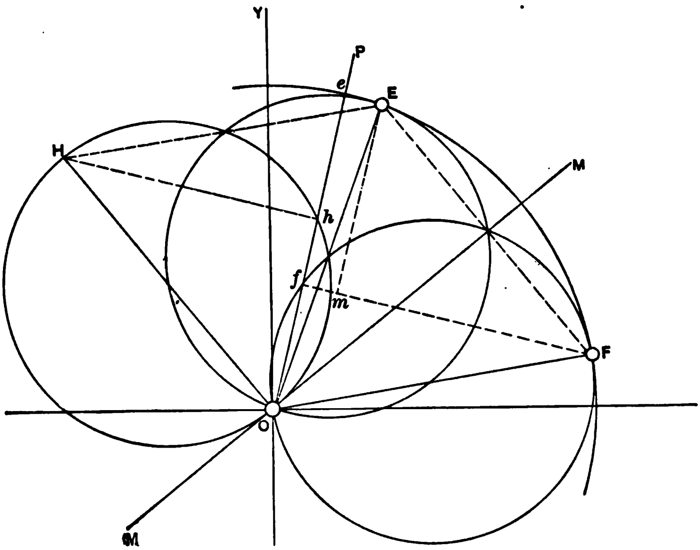


Fig. 59. Zeuner Diagram for Meyer Valve.

and *OF* that of the cut-off eccentric. These two radii are usually equal as a matter of convenience, while the angular advance *YOF* is about 90 deg. Draw two valve circles on *OE* and *OF* as diameters and draw any crank line *OP*. *Oe* will be the travel of the main valve and *Of* that of the cut-off valve at the same instant. The center lines of the two valves will be the distance *ef* apart and this will be called the relative travel. Join *EF*, draw *OH* equal and parallel to *EF* and construct a third valve circle (called the

relative circle) on OH. It can be easily proved from the figure that $Oh = ef$. For drawing straight lines Ee, Ff and Hh, we have right angles at e, f and h. Drop the perpendicular Em on Ff. $Em = ef$, but by similar and equal triangles, OHh and EFm, $Em = Oh$. Therefore $Oh = ef =$ relative travel.

We need not then consider the two first valve circles, but find the intercept Oh of any crank position on the relative circle and this will be the relative travel of the cut-off valve.

Referring now to Fig. 60, we will use the following notation:

$w =$ width of valve port in main valve.

$b =$ width of cut-off block.

$c =$ clearance of cut-off block when in mid-position on main valve.

$x =$ room for adjustment.

Mid-position of cut-off valve as shown in the figure would correspond to no travel on the relative valve circle; or the crank position tangent to the circle as OM (Fig. 59).

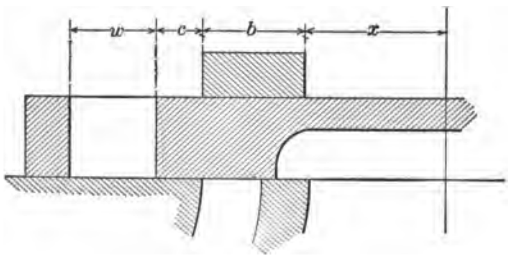


Fig. 60.

The block must travel from this position a distance $c + w$ in order to cut off steam from main valve. It will continue its travel beyond this until total travel $= OH$ (Fig. 59) and will lap over the port an amount $= OH - (c + w)$.

To prevent opening on the rear side b must at least equal $OH - (c + w) + w = OH - c$.

The block will now return and will reopen the port when travel is again equal to $c + w$. The cut-off and admission should precede those of the main valve.

An example will be solved to show the application of the diagram in designing a double valve:

Assume the following data:

Width of port $= w = 1.25$ in.
Width of bridge $= b = 2$ in.
Angle of lead $= 5$ deg.
Main cut-off at 0.75 stroke.
Block cut-off from 0.25 stroke upwards.
Compression at 0.875 stroke.

The design of the main valve is in all respects similar to that in Art. 51 and need not be described in detail. The results are as shown in Fig. 61.

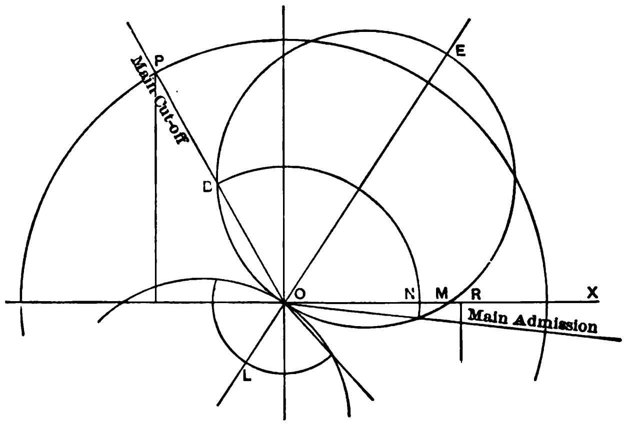


Fig. 61. Design of Main Valve (Meyer Valve).

The cut-off eccentric OF (Fig. 62) will have the same radius as the main eccentric OE and will be located with an angular advance $= 90$ deg. This is appropriate for a reversing engine, for when the main eccentric is shifted to OE', OF will occupy a similar relative position.

OH is now found as in Fig. 59 and the relative circle drawn. OQ is the crank position for the auxiliary or block cut-off and this intersects the circle at K so that $OK = c + w$. If $OK > w$ the clearance c is positive as in Fig. 60, but if $OK < w$ the clearance

is negative and the block projects over the edge of port the amount $w - OK$.

The valve again admits steam at G. If this is later than the main admission the design is at fault.

The width of block b must at least equal $OH - c$, as before explained.

To adjust for a later cut-off the blocks must be moved together, increasing the distance c.

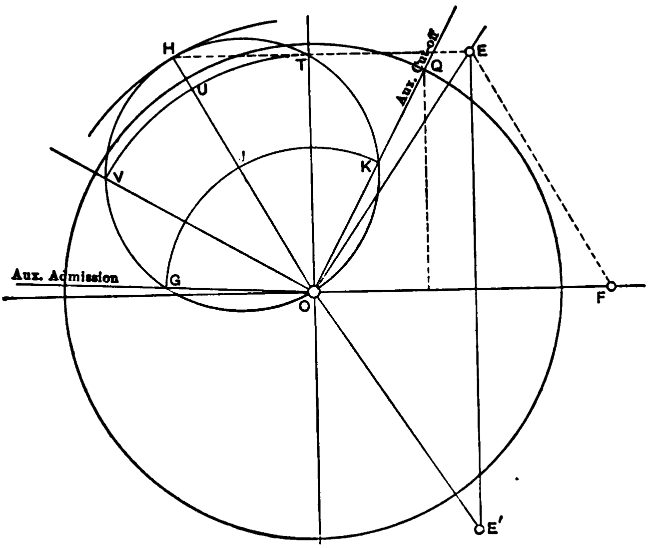


Fig. 62. Design of Cut-off Valve (Meyer Valve).

When $w + c = OT$, cut-off will occur at half stroke and admission at V. HJ in the first case and HU in the second show the amount of the valve laps over the port $= OH - (c + w)$. When $w + c = OH$, the valve does not overlap but closes and opens again instantly. This is the limit of useful adjustment and we have $x = JH$. (See Fig. 60.) If the blocks are moved still nearer together they will not close the ports and the device becomes inoperative. Since the main valve closes at OP (Fig. 61), it is desirable that OH and OP should make about the same

angle with OX. In such case the whole range of adjustment from OQ to OP is secured without any gap.

If the valve motion is to be used on a non-reversing engine or one which usually runs forward, it is better to increase the angular advance of OF slightly. This increases OH and OK and gives larger port openings and an earlier admission. Fig. 63 shows the complete valve as drawn to scale and in true position for the crank position OQ, Fig. 62.

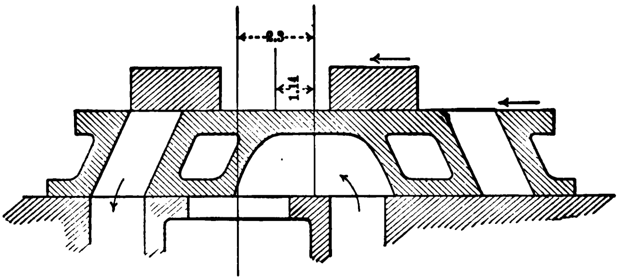


Fig. 63. Meyer Valve Designed in Figs. 61 and 62.

The values obtained in Figs. 61 and 62 are as follows:

Angular advance of main eccentric $= a = 32$ deg. 40 min.
Radius of main eccentric $= r = 2.3$ in.
Steam lap, main valve $= l = 1.05$ in.
Exhaust lap, main valve $= e = 0.375$ in.
Steam lead, main valve $= d = 0.18$ in.
Radius of cut-off eccentric $= r^1 = 2.2$ in.
Width of cut-off plates $= b = 2.3$ in.
Adjustment cut-off plates $= x = 1.05$ in.

Those valve motions which are dependent upon the action of the governor for their operation will be discussed in the chapter on governors.

Problems.

1. Design a common slide valve to meet the following conditions:

Port width $= w = 1.5$ in.
Bridge width $= b = 2$ in.

Exhaust lap $= e = 0.05$ in.
Average cut-off at 0.7 stroke.
Angle of lead $= 6$ deg.
Determine:
(a) Angular advance of eccentric.
(b) Radius of eccentric.
(c) Steam lap and lead.
(d) Average compression and release in parts of stroke.
Draw Zeuner diagram full size and valve half size.

2. Design a valve for conditions given in Problem 1, so that the cut-off may be at exactly 0.7 stroke in both strokes, assuming the connecting rod to be 5 cranks in length.

Determine steam laps and leads on head and crank ends of valve and show what lengthening of valve stem in Problem 1 would have nearly the same effect.

3. Show what effect the lengthening of the valve stem in Problem 1 would have on the times of compression and release in the two strokes.

Determine this exactly by construction on the diagram.

4. Design a cut-off or Meyer valve, using the conditions of Problem 1 for the main valve, and arranging the blocks to cut off from 0.2 to 0.7 stroke. Make the radius of the cut-off eccentric the same as the main radius and give it an angular advance of 100 deg.

Determine:

Width of blocks $= b$.
Clearance for either cut-off $= c$.
Amount of adjustment $= x$.

Draw valves in true position for crank angle $= 0$.

CHAPTER VI.

INDICATORS AND INDICATOR DIAGRAMS.

63. The Indicator.—The steam engine indicator is an instrument for recording simultaneously the motion of the piston and the variation of the steam pressure in the cylinder of an engine. Fig. 64 illustrates one of the best known of these instru-

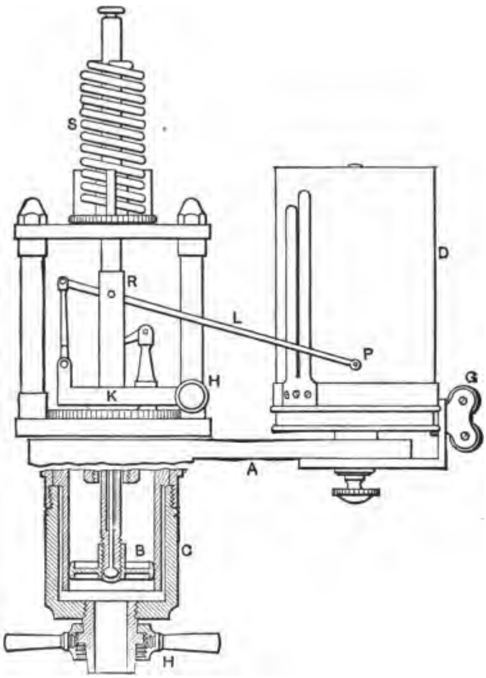


Fig. 64. Steam Engine Indicator.

ments and will serve for the purposes of description, since the principle of all indicators is the same.

The cylinder *C* of the indicator is attached by a screw union *H* to a plug-cock communicating with one end of the engine cylinder, so that steam has free access to the under side of the piston *B*,

whenever the cock is open. This piston moves vertically in the cylinder and is constrained by a helical spring *S*. The spring is fastened at the lower end to the frame of the instrument and at the upper end to an extension of the piston rod.

In most indicators the spring is inside the cylinder and exposed to steam heat. The outside spring shown is comparatively free from changes of temperature.

The upper end of the piston is exposed to atmospheric pressure and when steam is admitted underneath, the piston and its rod *R* rise an amount proportionate to the steam pressure.

The rod *R* is pivoted to a parallel motion carrying the lever *L* and the pencil point *P*. The motion shown in the figure is such that the point *P* travels five times as far as *R* and in a vertical straight line. The whole pencil movement is attached to the sleeve *K*, which can turn freely on a vertical axis, and thus bring the pencil to or from the paper.

The paper is attached by clips to the drum *D* turning on a vertical spindle. The drum is moved by a cord, passing over guide pulleys and connecting in some way with the crosshead or piston of the engine. A helical spring inside the drum brings it back as the cord slackens. The guide pulleys *G* are so arranged on swivels that the cord can be led off at any angle.

When the indicator is in operation the vertical movement of the pencil records the steam pressures and the horizontal movement of the paper shows the motion of the piston of the engine. The combination of these two motions gives the indicator diagram.

64. Scale of Pressures.—The piston of the indicator usually has an area of 0.5 sq. in., but when the indicator is used on gas engines a piston of half this size is frequently substituted on account of the higher pressures.

The parallel motion multiplies the movement, we will say, in the ratio of 5 to 1. The spring will be marked according to the scale of the pencil motion. For example, a 60 spring is one which will allow the pencil to move 1 inch for a pressure of 60 lb. per sq. in. The motion of the spring itself will be one fifth of this or at the rate of 1 in. for 300 lb. per sq. in. As the area of the piston is one half of a square inch the total pressure will be one half the gage pressure or 150 lb. for each inch of spring motion.

Such stiff springs are more satisfactory in action at high speeds than are flexible ones.

65. Reducing Motions.—The usual length of an indicator diagram is from 3 to 4 in. and this is the necessary motion of the cord. The length of stroke of steam engines varies from six inches to as many feet and this makes it necessary to interpose some mechanism for reducing the motion, between the crosshead and the indicator. The character of the motion must not be changed by the reduction and great care must be used in the selection of a correct mechanism.

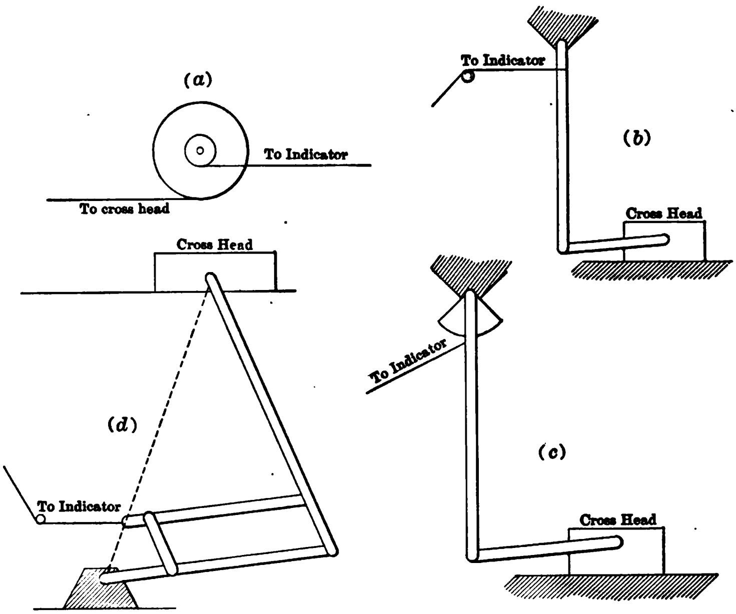


Fig. 65. Reducing Motions.

Fig. 65 illustrates a few of the more common devices. (a) is the drum motion, consisting of two pulleys of different diameters for the cords leading from the crosshead and to the indicator. The cord pulls the drum around and a spiral spring on the axis turns it back. When used at high speeds this device is apt to give a jerky motion, due to the inertia of the drum.

The two pendulum motions (b) and (c) are perhaps more

used than any other on account of their simplicity and the ease with which they can be installed. Neither of them gives an entirely correct motion, because the character of the connections at top and bottom is not the same, but if the lower link is of considerable length the error is not a serious one. (c) is superior to (b) in that no guide pulley is required. Long cords and numerous guide pulleys are objectionable and will cause inaccuracies in the diagrams. (d) is an example of a pantagraph which will give an exact motion to the indicator if the joints are well fitted and the cord is not too long.

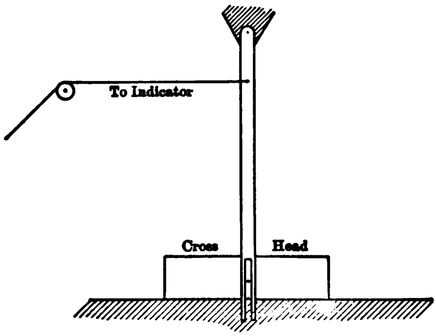


Fig. 66. Incorrect Motion.

No device should be used in which the lengths of any of the levers change by reason of sliding connections. Fig. 66 illustrates a common form of connection which has this fault. If the slot were in the crosshead instead of the lever, the motion would be nearly correct.

The cord should always lead off the lever on the plane of motion, and in devices which have no circular arc the cord must be parallel to the line of stroke as in (b) or (d), Fig. 65.

66. The Use of the Indicator.—The following simple rules are general in their character and will help any one to get correct indicator diagrams:

(1) Put the indicator together in accordance with the directions given in the handbook which accompanies it. Use a spring having a scale about two thirds the maximum steam pressure.

(2) Use one indicator on each end of cylinder if possible. If not, use one indicator on a three-way cock communicating with either end of cylinder.

(3) Blow off cylinder cocks thoroughly to remove all sediment and scale from pipes before attaching indicators.

(4) Set indicators and guide pulleys so that cords lead correctly from reducing motion.

(5) Adjust length of cord so that diagram shall come midway of the length of card.

(6) Adjust pencil movement so as to give sufficient pressure between pencil and paper.

(7) Take several preliminary diagrams to test apparatus and readjust if necessary.

(8) Take diagrams in order thus:

Hook up cord, starting indicator.

Open indicator cocks slowly.

Press pencils on paper for two or more revolutions.

Close indicator cocks.

Press pencils again to take atmospheric lines.

Unhook cords.

Remove cards and mark on each the number in series, the end of cylinder from which taken, the scale of spring, the speed of the engine and the boiler pressure.

67. Peculiarities of Diagrams.—The indicator may be used on an engine to determine if the valve is correctly set or to ascertain the indicated horse power. In the first case the scale of the diagram is of no importance and its shape alone need be considered. The main characteristics of the diagram have been shown in Figs. 6 and 30, but certain peculiarities of individual diagrams need to be noticed.

A slope of the admission line from right to left as at *DE*, Fig. 67, indicates excessive lead, while a slope from left to right as at the dotted line *De* shows a deficiency of lead. The eccentric should be turned forward or back to correct this until the admission line becomes vertical. The zigzag line at *E* is due to the vibration of the indicator spring and the high speed and may sometimes be eliminated by the use of a stiffer spring. The

gradual slope of the admission line from *e* to *A* is due to cylinder condensation and leakage. It sometimes indicates wire drawing of the steam due to insufficient port opening for the speed used.

The point *B* shows a late release or too much exhaust lap, while at *b* the release is too early, as shown by dotted lines. The

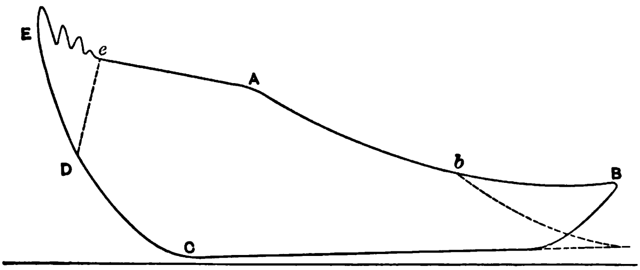


Fig. 67. Diagram Characteristics.

exhaust opening should be so timed as to round the toe of the card equally above and below. (See Fig. 6.)

A hook or point at the end of the compression, as *D*, Fig. 68, is usually caused by some leak in the cylinder, as for instance an

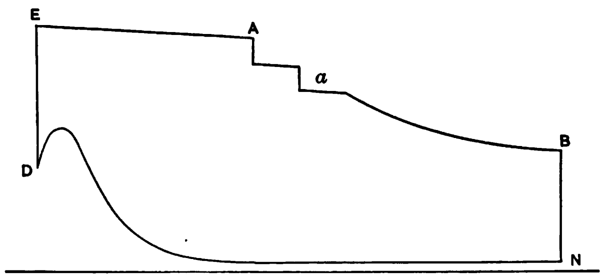


Fig. 68. Diagram Characteristics.

open drip valve, but may be due to a partial closing of the opening to the indicator pipe by the piston of the engine. A straight horizontal admission line like *EA*, Fig. 68, is suspicious, as usually

showing some stoppage or sticking of the indicator piston. Rectangular steps on the expansion line, like *Aa,* are further evidence of friction in the indicator mechanism.

Sharp corners and a vertical line at either end of the card, as *BN,* mean that the indicator cord is too short or too long and allows the drum to strike the stop at one end of the stroke.

Fig. 69 shows a pair of diagrams taken from the head and crank ends of an engine whose valve is out of adjustment, the stem being too short. This has the effect of keeping the steam port open too long on the head end while the exhaust port is

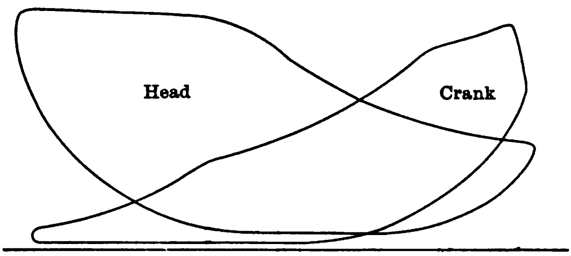


Fig. 69. Valve Out of Adjustment.

cramped. On the crank end the steam port does not open enough to admit steam at full pressure and cut-off and release occur too soon.

Numerous other examples might be given, but those mentioned are some of the most important. A valve may be set never so carefully by measurement, the unequal expansion caused by the hot steam will sometimes change its position, and the indicator must be consulted for the final adjustment.

68. Horse Power from Diagrams.—It has already been shown in the introduction that the indicated horse power of an engine is given by the formula

$$\text{I.h.p.} = \frac{PLAN}{33{,}000}$$

where P is the mean effective pressure on the piston as determined from the indicator diagram. The area of the diagram in

square inches divided by its length in inches gives the mean height in inches, and this last multiplied by the scale of the spring gives P.

The area of the diagram should be determined by a planimeter. Approximate methods for getting this area, such as dividing it into triangles or trapezoids, weighing a template of the diagram, etc., are unsatisfactory and should only be used in case of necessity.

To determine the true length of the diagram lines should be drawn perpendicular to the atmospheric line and tangent to the ends of the diagram. Measuring the atmospheric line is unsafe, as it may be longer or shorter than the diagram. The area A means the net area of the piston, the area of piston rod being deducted. This deduction must always be made on the crank end of the cylinder, while in tandem engines, pumps, air compressors, etc., there may be several rods of different sizes for the various cylinders.

The value of N depends upon circumstances. If the engine is single acting, N represents the number of complete revolutions. If double acting with diagrams from both ends of the cylinder, N still means the number of revolutions, each diagram is calculated separately and the results added together. If the engine is double acting and the horse power is to be calculated from one diagram, N must be taken as the number of strokes. With gas engines, single or double, two-cycle, four-cycle, hit-or-miss, etc., the value of N depends upon the conditions present.

The following rule is general and will apply to any combination of circumstances. *Each diagram represents a definite cycle of operations. N should always be taken as the number of cycles per minute of the kind shown by the diagram under consideration.*

Example.—An engine has a cylinder 17 in. in diameter by 24 in. stroke and makes 120 r.p.m. There is one piston rod 2.5 in. in diameter. Indicator diagrams have been taken with 60 lb. springs and measure as follows:

	Head End.	Crank End.
Area	2.4 sq. in.	2.64 sq. in.
Length	3.2 in.	3.2 in.

It is required to find the indicated horse power. The mean effective pressure of the head end card is:

$$P = \frac{2.4 \times 60}{3.2} = 45 \text{ lb.}$$

and of the crank end:

$$P = \frac{2.64 \times 60}{3.2} = 49.5 \text{ lb.}$$

The area of the head end of the piston is 227 sq. in. L is equal to 2 ft. in our formula and N is 120, since we have separate cards for each end. The horse power of the head end is then:

$$\text{H.p.h.} = \frac{45 \times 2 \times 227 \times 120}{33{,}000} = 74.3$$

and for the crank end:

$$\text{H.p.c.} = \frac{49.5 \times 2 \times 222.1 \times 120}{33{,}000} = 80$$

The total indicated horse power of the engine is:

$$\text{I.h.p.} = 80 + 74.3 = 154.3.$$

In taking a number of diagrams from one engine it is sometimes convenient to combine the constants in the horse power formula and obtain one constant for each end of the cylinder. If we take

$$K = \frac{LA}{33{,}000}$$

the formula will then read: h.p. $= KPN.$

In the example just used

$$K = \frac{2 \times 227}{33{,}000} = 0.01376 \text{ for the head end of the cylinder and}$$

$$K = \frac{2 \times 222.1}{33{,}000} = 0.01346 \text{ for the crank end.}$$

These so-called engine constants serve to shorten the work of calculation when a number of sets of diagrams are taken from the same engine.

Problem.

A compound tandem pumping engine has the following dimensions: Stroke, 48 in.; diameter of steam pistons—high pressure 24 in., low pressure 48 in.; diameter of water plungers, each 20 in.; diameter of rods between steam and water cylinders, each 3 in. The engine makes 12 complete cycles per minute and the water pressure in the main is 65 lb. per sq. in.

Indicator cards taken show the following dimensions:

	Area	Length	Scale
H.P. head	2.12	3.6	80
H.P. crank	2.23	3.6	80
L.P. head	2.54	4.1	20
L.P. crank	2.48	4.1	20

By head end is meant the end of the steam cylinder farthest from the water cylinder.

Make a table showing the total mean effective pressure on each side of each piston or plunger, the horse power developed in each cylinder and the loss of power by friction.

CHAPTER VII.

COMPOUND ENGINES.

69. Definitions.—A compound engine is one in which the steam, after doing work in one cylinder, passes into one or more other cylinders in succession and does further work by expansion.

When the term compound is used alone, it generally means an engine having two cylinders. A three-stage compound is called a triple expansion engine, a four-stage is called a quadruple expansion engine, and so on.

Compound engines are further classified according to the arrangement of the cylinders. The first cylinder in the combination is called high pressure (HP) and the last low pressure (LP). The intervening cylinders, if there are such, are called intermediates (IP). If the (HP) and the (LP) pistons are on one rod the engine is said to be a tandem. Vertical tandem engines are sometimes called steeple engines.

If the two cylinders are arranged in parallel instead of in series, forming two engines working side by side on two cranks attached to the same shaft, the engine is said to be a cross compound.

A tandem engine has but one frame and one set of moving parts and costs but little more than a simple engine.

A cross compound, on the other hand, is practically a double engine with all the parts duplicated save the shaft and fly wheel.

With the triple expansion several arrangements are possible. three cylinders tandem, three cylinders in parallel, two tandem and one at the side, etc.

The low-pressure cylinder in any compound may consist of one large cylinder or two smaller ones.

In any ordinary cross-compound engine the (HP) crank is usually set from 45 deg. to 90 deg. ahead of the (LP) crank, but this is not a necessity.

Where the cranks of a compound engine are set at some angle other than 0 or 180 deg. it is necessary to provide a receiver as a tarrying place for the steam on its way from one cylinder to the other. For instance, if the cranks are at right angles the steam

will be released from the (HP) cylinder more than half a stroke before it is admitted to the (LP) cylinder.

Triple expansion engines usually have three cranks set 120 deg. apart.

70. Advantages.—There are several reasons why compound engines are better than simple ones.

(1) Saturated steam cannot be expanded more than four or five times in one cylinder, without too great a range of temperature and consequent condensation. High pressure steam cannot therefore be used economically in a simple engine, since the terminal pressure would be too high. If the high pressure steam is introduced into a small cylinder and expanded there, and is then carried at a lower pressure into the main cylinder, where it is further expanded to the desired terminal pressure, the temperature range will not be excessive in either cylinder, while the expansive force of the steam will be fully utilized. Furthermore the hotter steam comes in contact only with the smaller areas of the (HP) cylinders, so that there is less surface for condensation.

(2) A simple engine having an early cut-off and therefore a large ratio between the initial and terminal pressures is subjected to great fluctuations of stress. Such an engine of a given power will need a stronger frame and running parts and a heavier fly wheel than if the pressure range were less. A compound engine can have the same ratio of expansion with much less fluctuation of energy in each cylinder. This will be better understood by referring to Chapter IX.

(3) Where the cross-compound type of engine is employed and the cranks are set 90 deg. apart, the turning effort on the crank is more uniform and the engine runs more steadily than is the case with a single crank engine.

71. Arrangement of Cylinders.—A distinction should be made between a compound engine and a duplex engine. The latter term is used to designate an engine having two simple or high-pressure cylinders side by side and driving one crank shaft. The ordinary locomotive is an example of this, the cylinders being of the same size and using the same quality of steam. When one cylinder is made smaller and receives the steam first, passing it

over to the other, the locomotive becomes a two-cylinder compound.

Figs. 70 and 71 show the more common arrangements ot cylinders and cranks in compound engines. The feathered arrows indicate the directions of motion of the pistons. The bare arrows indicate the paths of the steam from cylinder to cylinder. 1 is the ordinary tandem with two cylinders. 2 and 3 are cross compounds, but with different crank arrangements.

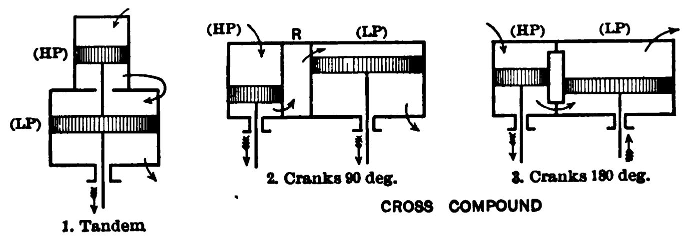


Fig. 70. Compound Cylinders.

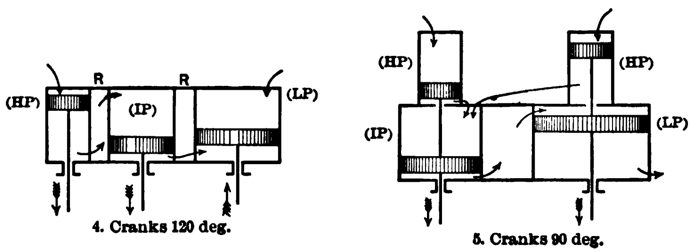


Fig. 71. Triple-Expansion Cylinders.

When the cranks are at 90 deg. it is necessary to employ an intermediate receiver (marked R in the figure) to hold the steam during the interval between the (HP) release and the (LP) admission.

When the cranks are at 180 deg. as shown in 3 the steam can pass directly across from one cylinder to the other and a receiver is not necessary.

At 4 is shown the usual arrangement for triple expansion engines, there being three cylinders and two receivers. In 5 four

cylinders are used, there being two high-pressure cylinders arranged in tandem with the intermediate and low-pressure respectively, and working on cranks at 90 deg. This arrangement is more compact than the other, but not so economical.

72. **Sizes of Cylinders.**—In determining the proportions of the cylinders of a compound engine it is convenient to first consider a single cylinder which shall do the same work with the same quality and quantity of steam and the same rate of expansion.

Let *ABCDE,* Fig. 72, represent the indicator diagram of such a single cylinder with a high rate of expansion, clearance and

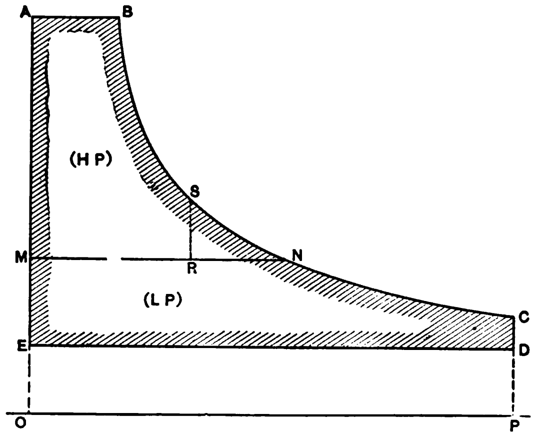


Fig. 72. Ideal Diagram of Work of Compound Engine Done in Single Cylinder.

compression being neglected. Let the area be divided into two approximately equal parts by the horizontal line *MN.* Then it is evident that if we neglect minor losses due to friction and condensation, the area *ABNM* may represent the indicator diagram of a (HP) cylinder having a volume $= MN$, an initial pressure $=$ *OA* and a back pressure in the receiver $= OM$.

Also the area *MNCDE* may represent the diagram of a (LP) cylinder having the same volume *ED* as the original single cylinder, an initial pressure from the receiver $= OM$ and a cut-off at *N.*

Since the volume of the (HP) cylinder $= MN$, the ratio of expansion in that cylinder $= \frac{MN}{AB} = r$. The ratio of the two cylinder volumes $= \frac{ED}{MN} = R$ and the product of these two ratios gives the total rate of expansion $= \frac{ED}{AB}$. In other words, the rate of expansion depends solely upon the volumes of the incoming and outgoing steam and not upon any intermediate processes.

In order that the work may be equally divided between the two cylinders the areas and mean pressures of the two diagrams must be equal. It is also desirable that the initial pressures on the two pistons may be equal, so that the margin of safety may be the same on both sides of the engine. In other words, $AM \times$ area of (HP) piston $= ME \times$ area of (LP) piston.

The stroke is usually the same in both cylinders and the areas of the pistons have the same ratio as the volumes of the cylinders, or in symbols

$$\frac{A}{a} = \frac{V}{v} = R$$

To insure equal initial pressures:

$$\frac{AM}{ME} = \frac{A}{a} = \frac{V}{v} = \frac{ED}{MN}$$

It is frequently found that when the line MN is so located as to make the areas equal above and below, the initial pressures will not be equal, that of the (HP) cylinder being too large.

The volume of the (HP) cylinder may then be reduced to MR (Fig. 72), changing R to $\frac{ED}{MR}$ and causing a drop of pressure SR between the (HP) cylinder and the receiver. This change gives a loss of work due to the small area SRN and impairs the equality of mean pressures.

A slight lowering of the receiver pressure will restore this equality, but will disturb the ratio $\frac{AM}{ME}$. By changing MN and SR it is usually possible to adjust both initial and mean pressures as closely as is necessary for actual running.

It will be noticed that these changes do not affect the total ratio of expansion nor the initial and terminal pressures. Also that the only effect of changing the cut-off point N in the (LP) cylinder is to raise or lower the receiver pressure OM, such change not affecting the total ratio of expansion.

73. Diagram Factors.—In actual diagrams the receiver pressure is usually not constant. The intermittent inflow and outflow of steam cause fluctuations of pressure which are most noticeable with small receivers. (See Figs. 81 and 82.) The larger the receiver volume, the more nearly is MN a horizontal straight line.

In some other respects the actual diagram as shown by the shaded area in Fig. 73 fails to correspond to the ideal diagram.

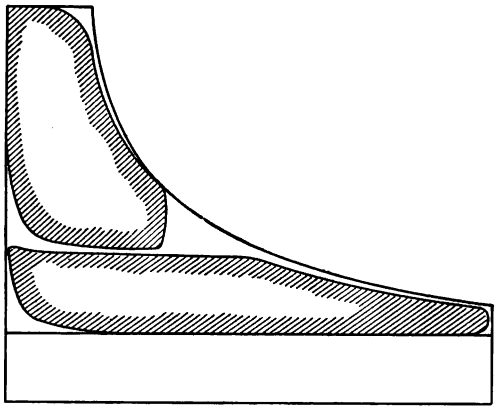


Fig. 73. Actual Diagram Corresponding to Fig. 72.

The corners of the actual diagrams are rounded by the friction or wire drawing of the steam passing through narrow ports and by the compression curves. The expansion lines diverge from the hyperbola and there is a distinct gap between the two diagrams, caused by receiver losses. It is customary to allow for these losses by the use of a *diagram factor*, determined by experiment and representing the ratio between the shaded area and the circumscribing diagram in Fig. 73. Professor Ripper gives this

factor as varying from 60 to 80 per cent for high speed and from 70 to 85 per cent for slower speed engines. The addition of a steam jacket may raise the factor to 90 per cent in the latter case.

In an unjacketed engine the terminal pressure is generally lower for the same ratio of expansion than that given by the ideal diagram. As it is difficult to predict this difference it will be neglected in the preliminary design of the cylinders. As originally taken by the indicator the (HP) and (LP) diagrams are generally of the same length, but with a different scale of pressures. In order to compare them with the ideal diagram it is necessary to redraw one or both of them, forming what is called a *combined diagram* (Fig. 73). In this diagram the pressure scales are the same and the lengths are in the same ratio as the cylinder volumes.

Such a diagram is necessary to a clear understanding of the principles of a compound engine, but is of little practical use otherwise, since all that needs to be known about the horse power and the valve adjustment can be found by a study of the separate diagrams, as taken by the indicator.

74. Example in Design.—In designing a compound engine, the sizes of the cylinders can be determined with a sufficient degree of accuracy from the ideal diagram, Fig. 72, by using a suitable diagram factor.

The terminal pressure may be safely assumed as from 10 to 12 lb. for condensing engines and from 20 to 25 lb. for non-condensing engines.

For boiler pressures of 100 lb. or less it would hardly pay to use more than one cylinder, while with steam at initial pressures ranging from 100 to 150 lb. absolute, two-cylinder compounds are efficient. For pressures higher than that last mentioned it would be advisable to use three or more cylinders.

No exact rule can be given for the ratio of cylinder volumes, as authorities differ. In general it may be said that the higher the initial pressure to be used, the larger this ratio will be.

For a similar reason non-condensing compounds would have a smaller cylinder ratio than condensing engines because of the higher terminal pressure. In ordinary practice this ratio varies from 3 to 1 to 4 to 1. Mr. Geo. I. Rockwood has built efficient compound engines with a ratio as high as 7 to 1.

Professor Ripper gives the following as representing English practice in three-cylinder compounds:

Boiler Pressure	(HP) vol.	(IP) vol.	(LP) vol.
140	1	2¼	6½
160	1	2⅜	7
180	1	2½	7½

To illustrate the use of the combined diagram in proportioning cylinders the following example will be solved:

Example.—Let it be required to find the proportions of cylinders for a two-cylinder compound engine of 200 h.p., using steam with an initial pressure of 130 lb. gage, a terminal pressure of 5 lb. gage and exhausting into the atmosphere.

Assuming the atmospheric pressure as 15 lb. and allowing 2 lb. for back pressure the diagram for a single cylinder will be as in Fig. 72, if clearance and condensation are neglected. The pressures will be: $OA = 145$ lb.; $PC = 20$ lb.; $OE = 17$ lb.

If the actual horse power is to be 200 and a diagram factor of 80 per cent is allowed, the horse power of the single diagram must be $200 \div 0.80 = 250$.

The total ratio of expansion

$$r' = \frac{145}{20} = 7.25.$$

The mean effective pressure in the single cylinder will be:

$$P = p_2\,(1 + \log_e r') - p_3 = 20\,(1 + \log_e 7.25) - 17 = 42.6.$$

Using the formula

$$\text{h.p.} = \frac{PLAN}{33{,}000}$$

we have

$$A = \frac{33{,}000\ (\text{h.p.})}{PLN}$$

Assume a piston speed $LN = 600$ ft. per min. (a very usual value).

$$A = \frac{33{,}000 \times 250}{42.6 \times 600} = 323 \text{ sq. in.}$$

corresponding to a diameter of a little over 20 in.

Assume the diameter of the (LP) cylinder as 20 in. and that of the (HP) cylinder as 12 in., which gives the ratio of cylinder volumes as $\frac{314}{113} = 2.78$.

The ratio of expansion in the (HP) cylinder will be

$$\frac{r'}{R} = \frac{7.25}{2.78} = 2.6$$

In Fig. 72, taking the volume AB as unity, we have the volumes: $AB = 1$; $MR = 2.6$; $ED = 7.25$.

It now remains to find the receiver pressure OM. If we assume the initial pressure on the two pistons as equal and call the receiver pressure x we shall have:

$$(145-x) \times (\text{HP}) \text{ volume} = (x-17) \times (\text{LP}) \text{ volume}$$
$$\text{or } (145-x) = 2.78\ (x-17)$$
$$\text{solving for } x$$
$$x = 51$$

This may be regarded as a trial value of x.

The mean effective pressure in the (HP) cylinder is:

$$P = \frac{145(1+\log_e 2.6)}{2.6} - 51 = 58 \text{ lb.}$$

The ratio of expansion in the (LP) cylinder is:

$$\frac{51}{20} = 2.55$$

and the mean effective pressure:

$$p' = \frac{51(1+\log_e 2.55)}{2.55} - 17 = 21.7 \text{ lb.}$$

The ratio of $\frac{p}{p'} = \frac{58}{21.7} = 2.67$ while the ratio of cylinder volumes as assumed $= 2.78$.

The horse power of the (HP) cylinder is accordingly less than that of the (LP) cylinder. If the receiver pressure is lowered to 50 lb. the values become: $p = 59$ lb.; $p' = 21.3$ lb.:

$$\frac{p}{p'} = \frac{59}{21.3} = 2.77$$

The work done in the two cylinders is thus practically the same, while the initial pressures are not sufficiently out of balance to cause any difficulty.

The horse power of the (HP) cylinder will now be:

$$\text{h.p.} = \frac{59 \times 113 \times 600}{33{,}000} = 121, \text{ and of the (LP) cylinder:}$$

$$\text{h.p.} = \frac{21.3 \times 314 \times 600}{33{,}000} = 121.5, \text{ a total of } 242.5.$$

The missing 7.5 h.p. is due in part to the small triangular area *SRN*.

The fall of pressure during admission, the rounding of corners at release and compression and the gap between the two diagrams as illustrated in Fig. 73 will further reduce the horse power to about the amount assumed at the start.

75. Variation in Load.—A well-designed compound engine running at a constant load is efficient and economical. When the load varies considerably, it is difficult to so regulate the compound engine as to maintain good efficiency. With more than two cylinders the difficulty increases and it is doubtful if triple engines should ever be used unless the load is nearly constant.

There are three ways in which the distribution of work between the two cylinders of a compound engine may be regulated: (a) By varying the (HP) cut-off; (b) by varying the (LP) cut-off; (c) by varying the initial pressure.

In studying these changes on a diagram it is necessary to remember that, once the engine is designed, the cylinder volumes are constant; i.e. in Fig. 72, *MR* and *ED* cannot be changed in length.

(a) In Fig. 74 let *ABSRM* and *MNCDE* be the indicator diagrams of the (HP) and (LP) cylinders respectively at nor-

mal load. If the load diminishes and speed increases, the action of the governor may change the cut-off point from *B* to *b*, and *bc* will be the new expansion line.

If there is no change in the (LP) cut-off, *n* will be vertically under *N* and the receiver pressure will fall from *OM* to *Om*. The new diagrams will be *Absrm* and *mncDE*. It is evident that the (HP) diagram has gained nearly as much as it has lost, while the (LP) diagram is much smaller than before.

A compound engine which is regulated by the (HP) cut-off alone will only work efficiently at the normal load. When the engine is overloaded the (LP) cylinder will do too much work, while with a light load the most of the work will come on the (HP) cylinder.

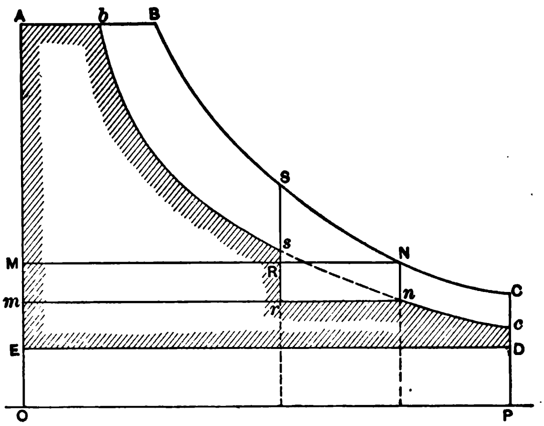


Fig. 74. Variable Load—Cut-off changed in HP Cylinder.

(b) A change of the cut-off point *N* in the (LP) cylinder merely serves to raise or lower the receiver line *MN* without changing the expansion line. If the cut-off is later the receiver pressure is lower and less work is done in the (LP) cylinder. Just the reverse is true when the cut-off is earlier. The only reason for changing the (LP) cut-off is to divide the work equally between the two cylinders.

From what has been shown in (a), it is apparent that the (LP) cut-off should be changed at the same time and in the same direc-

tion as that of the (HP) cylinder. In Fig. 75 the (HP) cut-off is moved back from B to b and at the same time the (LP) cut-off is shifted from N to n. The equality of the (HP) and (LP) areas is thus maintained, the receiver pressure kept up and the loss from "drop" is reduced.

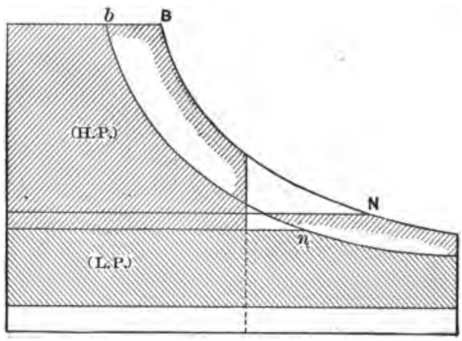


Fig. 75. Cut-off Changed in Both Cylinders.

Most two-cylinder compounds used for service under varying loads are equipped with variable cut-off mechanism on both cylinders. In order to get satisfactory regulation the (LP) cut-off should vary less rapidly than that on the (HP) cylinder. This is apparent from inspection of Fig. 75.

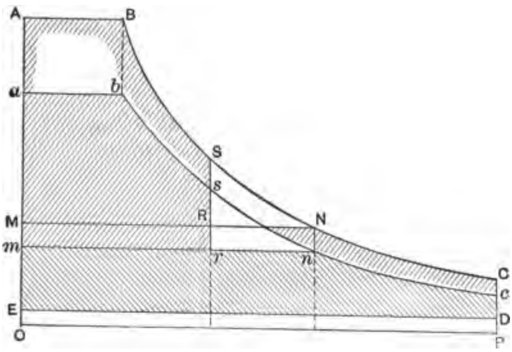


Fig. 76. Governing by Throttling.

(c) In Fig. 76 let $ABSRM$ and $MNCDE$ as before be the (HP) and (LP) diagrams at normal load. If the load diminishes

the initial pressure may be lowered to *ab* by hand throttle or by the action of a throttling governor. The expansion line is lowered to *bsnc* and intersects the vertical through N at a new point n. The receiver pressure is now $= Om$ and the equilibrium of areas is well maintained.

The ratio of expansion is not changed, which is another point in favor of this method. When there is a load change of any considerable duration it is probably more economical to meet it by a change of initial pressure than by a change of cut-off in either a simple or a compound engine. Where the fluctuations of load are rapid and violent the throttling method is hardly prompt enough for good regulation and the cut-off governor gives better satisfaction.

An arrangement which is common on triple expansion pumping engines having an approximately uniform load is as follows: The (HP) cut-off is regulated by a governor, controlling the speed for slight variations in the load. The (IP) cut-off can be regulated by hand for any permanent change of load. The (LP) cut-off is quite late in the stroke and is not changed.

76. Appearance of Actual Indicator Cards.—The diagrams as taken by the indicator are usually of the same length but with a different scale of spring for the two cylinders. Figs. 77-80 are facsimiles of diagrams taken from engines under actual running conditions, and show the relative scales of the springs used.

The diagrams in Fig 77 were taken from a horizontal cross-compound non-condensing engine, having Corliss valves on both cylinders, both actuated by one governor.

Fig. 78 shows diagrams from a condensing engine of another make, while Fig. 79 illustrates diagrams taken from a vertical compound engine of the marine type. There is no regulation of the cut-off on either cylinder and a throttling governor is used on the (HP) steam pipe. The diagrams from all three of these engines show an excess of work done by the (HP) cylinder.

The diagrams in Fig. 80 are from a vertical triple expansion pumping engine, having governor regulation on the (HP) valves and hand regulation on the (IP) valves. The (IP) cylinder is shown as not doing its share of the work. In determining the indicated horse power of a compound engine it is

sufficient to determine the horse power of each cylinder separately and add the results.

It will not be necessary to combine the diagrams by increasing the length of the (LP) diagram to correspond to its relative volume, as shown in Fig. 73. The combined diagram is useful

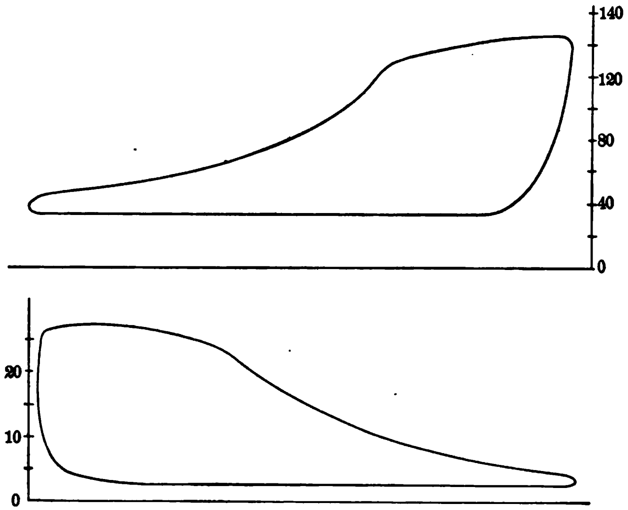


Fig. 77. Compound Engine Diagrams.

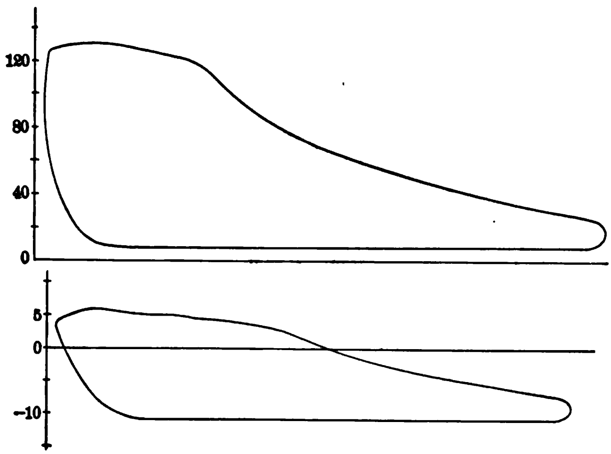


Fig. 78. Compound Engine Diagrams.

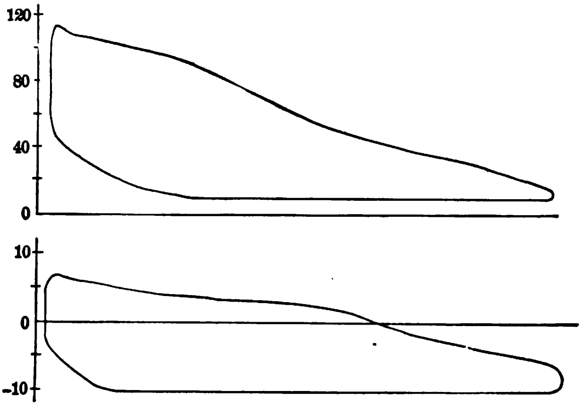


Fig. 79. Compound Engine Diagrams.

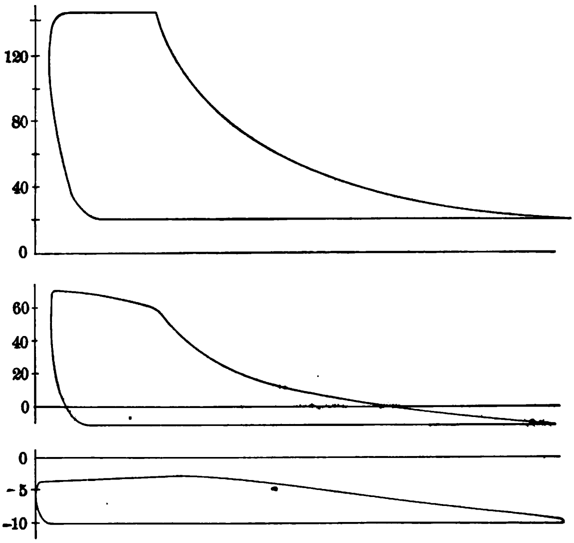


Fig. 80. Diagrams from Triple Expansion Engine.

for determining the proportions of the cylinders and the points of cut-off when designing new engines, but it has no particular value in testing actual engines and its importance has been much exaggerated.

77. Effect of a Small Receiver.—In the discussion of pressures and volumes so far the pressure in the receiver has been assumed to be constant or nearly so. An examination of the lines of pressure in Figs. 77-80 will show that this assumption is practically correct in some cases. It is evident that if the receiver volume is relatively large the pressure there will be but slightly affected by the inflow of steam on the one hand or the outflow on the other. In some engines the receiver is small, as may be the case when the strokes begin and end together. Fig. 81 shows the diagrams of a single-acting compound with cranks at 180 deg. The action is as follows:

Steam enters the head end of the (HP) cylinder from A to B and expands in that cylinder from B to C. Communication is

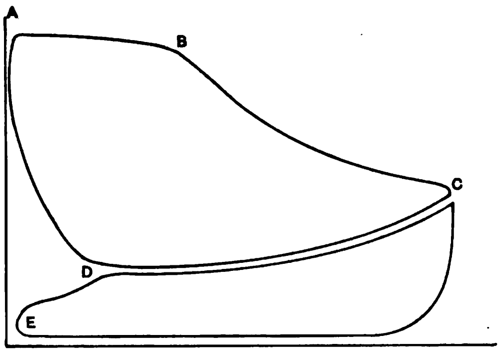


Fig. 81. Single Acting Compound Diagrams.

then opened with the head end of the (LP) cylinder through a short passage, and as the (HP) piston returns and the (LP) piston advances, steam expands from one cylinder to the other along the lines CD. The slight gap between the lines is due to friction in the ports. At D communication between the two cylinders is interrupted, the steam in (HP) cylinder is compressed to A and that in (LP) cylinder expands to E.

Fig. 82 on the other hand shows diagrams from a compound engine with cranks at right angles and having a small receiver. From D to E the steam is compressed from the (HP) cylinder into the closed receiver and the pressure rises. At E the valve

opens between the receiver and the (LP) cylinder, expansion begins as shown by the lines *EF* and *GH* and the pressure falls. At *H* on the (LP) diagram there is a sudden rise of pressure caused by the rush of steam from the other end of the (HP) cylinder into the receiver. *HI* is again expansion, while at *I* cut-off takes place and the expansion is still more rapid on account of the smaller volume present.

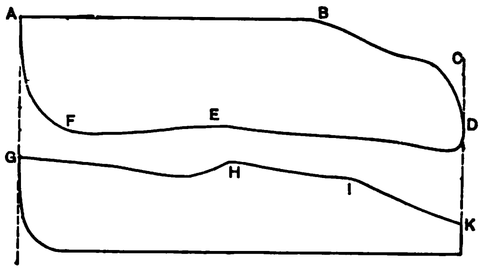


Fig. 82. Cranks at Right Angles—Small Receiver.

78. Chart of Volumes.—In order to understand clearly the shapes assumed by the pressure-volume lines in the indicator diagrams of a multi-cylinder engine, it will be necessary to construct a chart showing the volumes of steam present at different periods in one revolution of the engine.

One method of doing this is explained in Professor Ripper's work on "The Steam Engine" and will be used here with some slight modifications. In Fig. 83 the horizontal distances represent the volumes of the cylinders, the two clearances and the receiver, while the vertical divisions represent the time intervals for each half stroke. The sine curves show the changes of volume in each cylinder in equal times. The cranks are assumed to be at right angles and the curves therefore differ 90 deg. in phase.

The changes are as follows:

Time.	*Occurrence.*	*Volume of Steam.*
0	(HP) admission	c
1	(HP) cut-off	$\frac{3}{8}(HP)+c$
2	(HP) release	$(HP)+c+R$

Time.		*Occurrence.*	*Volume of Steam.*
3	(LP)	admission	$\frac{1}{2}(HP)+c+R+C$
4	(HP)	exhaust closure	$c+R+C+\frac{1}{2}(LP)$
5	(LP)	cut-off	$R+C+\frac{5}{8}(LP)$
6	(LP)	release	$C+\ (LP)$

The opening and closing of exhaust have been assumed at end of stroke in each cylinder to simplify the problem. The width of the shaded area at any point shows the volume of steam present at that time.

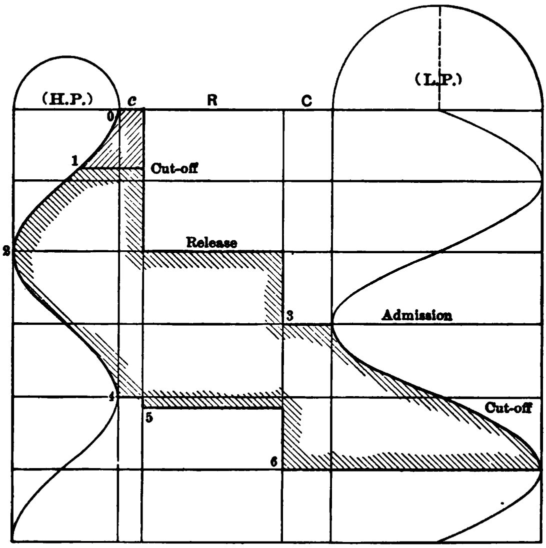


Fig. 83. Volume Chart.

To illustrate the application of the volume chart in the construction of indicator diagrams, the following data will be assumed:

Initial pressure = 130 lb.
Back pressure = 5 lb.
(HP) volume = 8 cu. ft.
(HP) clearance = c = 1 cu. ft.
Receiver volume = R = 12 cu. ft.
(LP) clearance = C = 2 cu. ft.

(LP) volume = 24 cu. ft.
(HP) cut-off at 3/8 stroke.
(LP) cut-off at 5/8 stroke.
Release at full stroke.
Compression neglected.

Hyperbolic expansion will be assumed, or

$$p\, v = p_1\, v_1$$

and reference will be made to Fig. 83. The pressures at the different times will all be calculated from the formula:

$$p = \frac{p_1\, v_1}{v}$$

The table shows the corresponding volumes and pressures at the points numbered in Figs. 83 and 84.

Time.	No.	Volume.	Pressure Formula.	Pressure Lb.
Cut-off (H.P.)	1	3 + 1	....	130.0
Before release (H. P.)	2 —	8 + 1	$\frac{130\times4}{9}$	57.8
Before release (L. P.)	6 —	24 + 2	$\frac{130\times4}{26}$	20.0
After cut-off (L. P.)	5 +	15 + 2	$\frac{20\times26}{17}$	30.6
Before cut-off (L. P.)	5 —	17 + 12	....	30.6 receiver.
After closure (H. P.)	4 +	12+2+12	$\frac{30.6\times29}{26}$	34.2
Before closure (H. P.)	4 —	26 + 1	....	34.2
After admission (L. P.)	3 +	4+1+12+2	$\frac{34.2\times27}{19}$	48.5
After release (H. P.)	2 +	8+1+12	$\frac{(57.8\times9)+(30.6\times12)}{21}$	42.2 receiver.
Before admission (L. P.)	3 —	4+1+12	$\frac{42.2\times22}{17}$	52.1 receiver.

Fig. 84 shows the two diagrams drawn to scale and located with the zeros of clearance volumes coinciding. The numbers correspond to those used in the table and in Fig. 83. Compression would round the heels of both diagrams and by changing the receiver pressure would slightly modify the lines along 2, 3, 4.

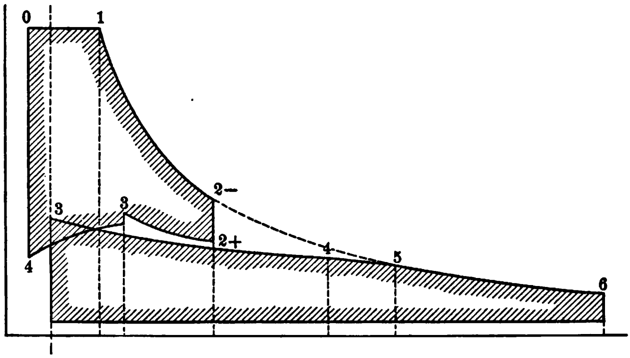


Fig. 84. Diagrams Corresponding to Fig. 83.

PROBLEMS.

1. Determine the proportions of the cylinder of a triple expansion engine under the following conditions:

Initial pressure, 180 lb. gage.
Terminal pressure, 12 lb. gage.
Back pressure, 2 lb. gage.
Indicated horse power = 600.
Piston speed = 720 ft. per min.
Diagram factor = 0.70.

Both the initial and the mean pressures are to be nearly equalized.

2. Draw a volume chart for the above engine, assuming the crank at 120 deg. and the volumes of the receivers as follows:

First receiver, (HP) volume × 2.
Second receiver, (IP) volume × 2.
Clearance 6 per cent in each cylinder.

3. Draw a combined diagram for the above engine, similar to that shown in Fig. 84 and calculate mean pressures in each cylinder, using the planimeter for measuring the areas.

CHAPTER VIII.

GOVERNORS.

79. Purpose and Classification.—The purpose of a steam engine governor is to regulate the average speed of the engine by adapting the supply of energy to the amount of work to be done. This purpose must be distinguished from that of a fly wheel, which is merely to regulate the speed of the engine during one cycle, controlling the minor fluctuations due to uneven effort or resistance. A fly wheel has no power to control the mean speed of an engine in the absence of a governor, but a heavy fly wheel materially assists a governor in speed regulation, by retarding the fluctuations of speed and giving the governor time to act.

A governor may regulate the steam in one of two ways, by changing the pressure, as in the throttling type, or by changing the quantity of steam admitted, as in so-called cut-off governors.

Governors may also be classified as *centrifugal* or *inertia,* according as centrifugal force or inertia is utilized for moving the mechanism which controls the admission of steam.

Centrifugal governors may be divided into *gravity* governors and *spring* governors. The first are usually of the pendulum type, and the action of the centrifugal force is balanced by that of gravity. In spring governors the movement of the weights is controlled by springs. So-called *shaft* governors belong either to the spring or to the inertia type.

80. Pendulum Governors.—The simplest form of pendulum governor, the so-called Watt governor, consists of a vertical spindle AD, Fig. 85, having pivoted at A two revolving arms AB. Each arm carries a ball B at the lower end, and is attached by a link LS to a sliding collar S on the spindle. As the spindle rotates, the arms and balls revolve with it and move up or down, according to the changes in speed, and by means of the links, slide the collar on the spindle.

The mechanism for controlling the admission of steam to the cylinder is actuated by the collar. As the speed increases, the

collar rises and cuts off steam, while a slowing down of the engine produces the contrary effect.

Considering the weight of one ball and neglecting that of the arm, link and collar, we have B, Fig. 85, in equilibrium under the action of its weight W and its centrifugal force C. The resultant T of these two forces is the tension on the arm and gives the direction of the latter.

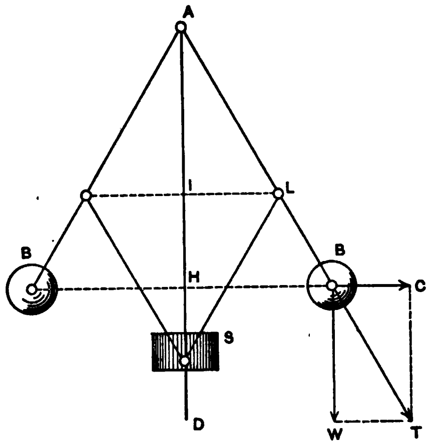


Fig. 85. Pendulum Governor.

The following notation will be adopted:

$W =$ weight of ball in pounds.
$C =$ centrifugal force of ball in pounds.
$r =$ radius of revolution in inches.
$= HB.$
$h =$ height of cone in inches.
$= AH.$
$N =$ number of revolutions per minute.
$v =$ velocity of ball in feet per second.

The usual expression for centrifugal force is:

$$C = \frac{Wv^2}{gR}$$

where R is the radius of revolution in feet, $= \dfrac{r}{12}$.

Substituting the value

$$v = \frac{2\pi NR}{60} = \frac{NR}{9.55}$$

and reducing,

$$C = \frac{WN^2R}{2933} = \frac{WN^2r}{35,200} \tag{19}$$

By similar triangles *AHB* and *TCB*,

$$\frac{h}{r} = \frac{W}{C}$$

$$h = \frac{Wr}{C} = \frac{35,200\ Wr}{WN^2r}$$

Reducing,

$$h = \frac{35,200}{N^2} \tag{20}$$

The distance *AH* of the planes of rotation of the balls below the apex of the cone, the so-called height of cone, is then inversely as the square of the speed and independent of any other dimension. The weight of ball and the length of arm have no effect upon the position of the balls. Several governors revolving about one spindle would all stand at the same height h when turning the same number of revolutions per minute, although they had different lengths of arm and different weights of balls.

81. Loaded Governors.—In the preceding discussion, the weights of the arms and links have been neglected, as they are usually small compared with the weight of the balls. The weight of the collar *S*, Fig. 85, may be considered, as it tends to keep the balls lower and retard the action of the governor.

In high-speed governors, this collar sometimes carries a heavy weight for the express purpose of holding the governor down and increasing the value of h. In Fig. 86 is shown a governor with comparatively light balls and a heavy weight *L* sliding on the spindle and connected with the balls by the links *BS*, having the same length as the arms *AB*.

The motion of *L* on the spindle will then be double the vertical motion of *B*, and consequently the vertical pressure on the two

balls will be double the load. The total vertical force acting on each ball is $L + W$, where L is the weight of the load. The centrifugal force C is not affected by the load since the latter is central.

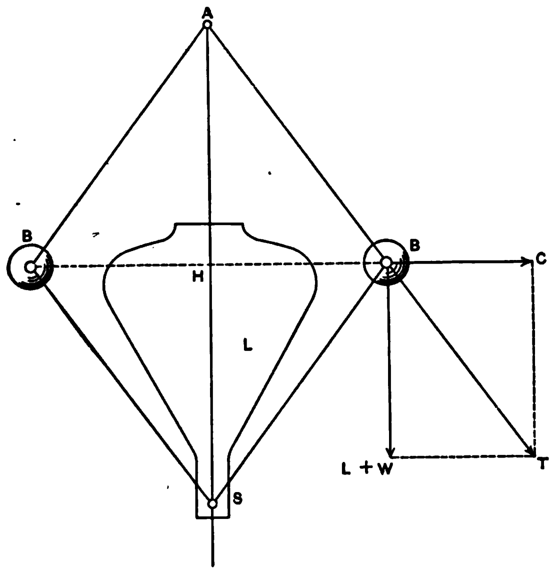


Fig. 86. Modified Pendulum Type Loaded.

The ratio of height to radius is now

$$\frac{h}{r} = \frac{L + W}{C}$$

Substituting the value of C from equation (19) and reducing,

$$h = \frac{L + W}{W}\left(\frac{35,200}{N^2}\right) \tag{21}$$

or the height of cone in Fig. 86 is greater than that of the unloaded governor by the ratio $\dfrac{L + W}{W}$.

If the load is placed on the collar of the Watt governor, as shown in Fig. 87, it will have only half the effect. The load

now moves with the same speed as the balls, and the equation for h is:

$$h = \frac{\frac{L}{2} + W}{W}\left(\frac{35{,}200}{N^2}\right)$$

$$= \left(\frac{L}{2W} + 1\right)\frac{35{,}200}{N^2} \qquad (22)$$

The effect of a center load on the governor in Fig. 85 would be intermediate between those already mentioned and would depend upon the relative motion of the point S.

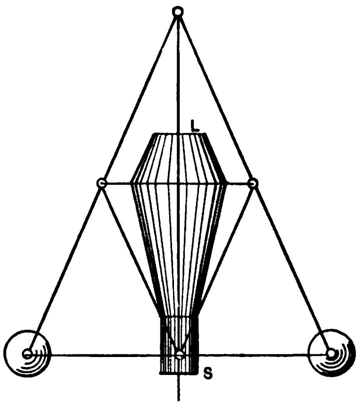


Fig. 87. Watt Type Loaded.

To illustrate the foregoing, an example will be worked from the following data:

A governor similar to that in Fig. 87 has balls weighing 2.25 lb. each and a collar S weighing 3 lb. Neglecting the weight of links, determine the value of h for a speed of 120 r.p.m: (1) when the collar S is unloaded, (2) when there is a load L of 40 lb. on the collar.

Substituting the value $W = 2.25$ and the values $L = 3$ and $L = 43$ in equation (22),

$$(1) \quad h = \left(\frac{3}{4.5} + 1\right)\frac{35{,}200}{120 \times 120} = 4.07 \text{ in.}$$

and

$$(2)\quad h=\left(\frac{43}{4.5}+1\right)\frac{35,200}{120\times 120}=25.8 \text{ in.}$$

The height of an unloaded governor at this speed would be only 2.44 in.

82. Fluctuation of Governor.—No governor, which depends upon centrifugal force for its operation, can maintain a uniform speed, since it requires a change of speed to render it operative. When the balls are in the lowest position, the throttle or the cut-off valve is open so as to supply enough steam for the maximum load, and when the balls are highest, just enough steam is admitted to run the engine unloaded.

Let the greatest allowable variation of speed between these limits $= \frac{1}{m}$ of the mean speed and let the corresponding change in height of cone $= k$.

Let N_1, N_2 and N be the maximum, minimum and mean speeds, respectively, expressed in revolutions per minute.

Let h_1, h_2 and h be the corresponding heights of cone. By definition,

$$N_1-N_2=\frac{N}{m} \tag{a}$$

$$N_1+N_2=2N \tag{b}$$

$$h_2-h_1=k \tag{c}$$

Solving in (a) and (b)

$$N_1=N\left(1+\frac{1}{2m}\right) \tag{d}$$

$$N_2=N\left(1-\frac{1}{2m}\right) \tag{e}$$

From equation (20)

$$h=\frac{35,200}{N^2}$$

By proportion,

$$h : h_1 : h_2=\frac{35,200}{N^2} : \frac{35,200}{N_1^2} : \frac{35,200}{N_2^2}$$

Substituting values from (d) and (e)

$$h : h_1 : h_2 = 1 : \frac{1}{\left(1+\frac{1}{2m}\right)^2} : \frac{1}{\left(1-\frac{1}{2m}\right)^2}$$

By composition,

$$h : h_2 - h_1 = 1 : \frac{\frac{2}{m}}{\left(1-\frac{1}{4m^2}\right)^2} \tag{23}$$

In most engines m is a comparatively large number, from 50 to 100, and therefore the denominator in (23) can be neglected. Inverting, we have approximately:

$$\frac{h_2 - h_1}{h} = \frac{k}{h} = \frac{2}{m} \tag{24}$$

Or in words, the rate of fluctuation of h is about double the rate of fluctuation of speed. Referring to the example in Art. 81, and assuming that a variation of speed of 2 per cent is allowed, the fluctuation in h will be about 4 per cent.

The change in height of cone would be 1.03 in. for the loaded governor in the preceding example and only 0.163 in. for the one with the 3 lb. collar. The latter would be comparatively useless as a regulator on account of the small movement.

This example shows the advantage of the loaded governor at high speeds. The raising of the heavy load also stores a certain amount of energy which is available for moving the valve mechanism in the event of a sudden increase of load, as in rolling mill or electric railway engines.

At comparatively low speeds, the height of cone and the range of the unloaded governor is sufficient, and if comparatively long arms and heavy balls are employed, the stored energy will be available.

For example, if the speed of the governor in the preceding examples was 50 r.p.m. instead of 120, h would be

$$\left(\frac{3}{4.5}+1\right)\frac{35,200}{50\times 50} = 23.5 \text{ in.}$$

for the governor without the 40-lb. load.

83. Sensitiveness.—A governor is said to be sensitive when a small variation of speed effects a considerable movement of the regulator. That governor is most sensitive in which a given value of $\frac{k}{h}$ will cause the greatest movement of the arms and balls. To illustrate this, consider the three arrangements shown in Fig. 88.

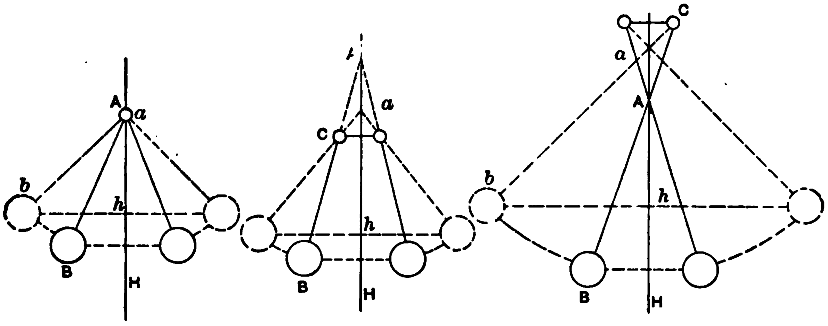


Fig. 88. Illustrating Sensitiveness of Governor.

In each case, AH is the height of cone at slower speed and ah the height at faster speed $= \frac{3}{4}$ of AH. Accordingly k is $\frac{1}{4}$ of AH, or approximately:

$$\frac{k}{h} = \frac{1}{4} \div \frac{7}{8} = \frac{2}{7} = \frac{2}{m} \text{ and } m = 7$$

In (1), the corresponding rise of balls is the same as k.

In (2), where the arms are attached to a cross-bar below the point A, the rise of balls is much less.

In (3), where the arms are crossed and attached to a bar above A, the rise of balls is much greater than h.

(2) is then the least sensitive of those shown and (3) the most so. According to this definition, the loaded governor is neither more nor less sensitive than the unloaded one, when the height of cone is the same. It is to be noted, however, that when h is large at a certain speed, k is also larger for a given change of speed, since $\frac{k}{h}$ is proportional to $\frac{1}{m}$.

Any governor is more effective when the arms make a small angle with the spindle. The two governors shown in Fig. 89 have the same length of arm and the same range of speed, but (1) is more efficient because h and k are both larger. This shows the reason for employing a center load at high speeds.

84. Stability and Isochronism.—A governor is said to be stable when it assumes a definite position for each particular speed and when a change of speed is necessary for a change of position. All the governors so far illustrated are stable.

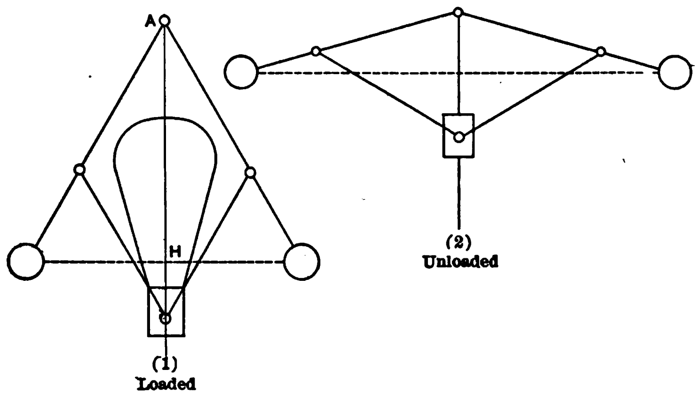


Fig. 89. Showing Reason for Loaded Type.

The condition necessary for stability in a pendulum governor is that the centrifugal force shall increase faster than the radius of revolution. By Art. 80,

$$\frac{h}{r}=\frac{W}{C} \text{ or } \frac{C}{r}=\frac{W}{h}$$

Since W is constant, as h decreases, $\frac{C}{r}$ must increase, and therefore C increases faster than r. This condition is due to the fact that the balls rise as the speed increases. If, as in Fig. 90, the balls were constrained to move out in a horizontal line, C would increase directly as r, and h would be constant. The governor would be in equilibrium only at the speed corresponding to h, and

at that speed the balls would be in equilibrium anywhere on the horizontal line, and the governor would be unstable.

If the arms were pivoted below the center so as to give a path to the balls indicated by the dotted lines, Fig. 90, gravity and the centrifugal force would act together to move the balls out and down, and the governor would not be in equilibrium at any speed.

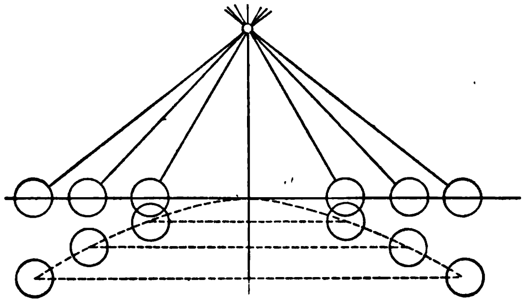


Fig. 90.

A governor which is in equilibrium at but one speed is said to be *isochronous*. The best example of this type is the parabolic governor, Fig. 91.

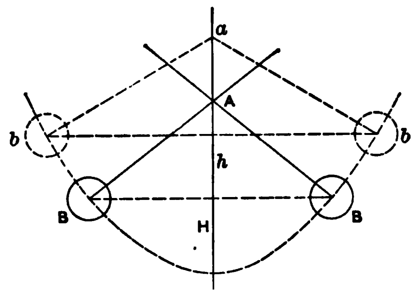


Fig. 91. Parabolic Type.

If the balls of a pendulum are constrained to move in the arc of a parabola, the value of h will be constant. For AH in the figure is the sub-normal of the curve, and by geometry, this is constant for any given parabola.

Such a governor is impracticable for mechanical reasons, but an approximation may be obtained by producing the two extreme positions of an arm BA and ba until they intersect at C and

using this as a point of suspension. See Fig. 88 (3). The circular arc Bb will be an approximation to the parabola, and the governor will be approximately isochronous.

If the point C is moved nearer to A, the governor becomes less sensitive and more stable. If the point C is moved too far away from the spindle, h may increase as the balls rise and the governor becomes useless.

An isochronous governor cannot be used successfully on an engine without some modifications. At the slightest increase in speed above the normal, the balls would rise to their highest position and cut off the steam. This would cause a sudden decrease of speed, and the governor would promptly fall to its lowest position and open wide the steam valve. The governor would thus augment the evil it was intended to correct. Such extreme fluctuations can be checked by the use of a spring or dashpot. See Art. 87.

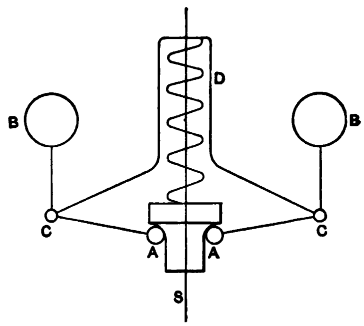


Fig. 92. Hartnell Governor.

85. Spring Governors.—The centrifugal force of governor balls may be balanced by a spring instead of by gravity, and in such a case, the axis of the governor is not necessarily vertical. Fig. 92 shows the principle of the Hartnell spring governor. The shell D revolves with the spindle and carries the two bell cranks ABC pivoted at C. The centrifugal force causes the balls B to fly outward and raise the ends A of the bell cranks. This motion raises the sleeve S against the pressure of the helical spring on the spindle. The sleeve in turn actuates the valve mechanism.

As the balls B move outward, the centrifugal force increases with the radius of revolution. The compression on the spring increases in a similar manner, and by suitably adjusting the initial pressure on the spring, the governor may be made nearly isochronous, and consequently, very sensitive. This is a characteristic of spring governors, which will be more fully explained in Art. 87.

86. Shaft Governors.—Most spring governors are mounted on a horizontal spindle, and when this coincides with the crankshaft of the engine, the governor becomes a *shaft governor.*

The ordinary pendulum governor must be driven by the engine through the medium of belts or gears, and there is always the chance of an accident due to breaking or slipping of the connecting mechanism. The shaft-governor usually forms a part of the fly wheel itself and turns when the engine does. Accidents with this type of governor are extremely rare and are due to sticking of the mechanism, through rust or poor lubrication.

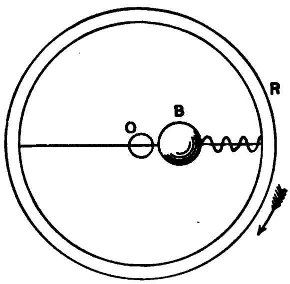


Fig. 93. Shaft Governor.

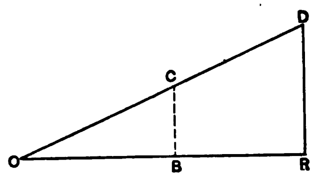


Fig. 94.

The simplest form of shaft governor is one having a weight or ball moving radially outward and opposed by a spring, as in Fig. 93. The performance of such a governor will depend entirely upon the location and scale of the spring. The following notation will be used:

Let $R =$ maximum radius of ball's path.
$r =$ any radius of ball's path.
$P =$ pressure on spring.
$C =$ centrifugal force of ball.

Suppose when the ball is at center that the spring just touches it with no initial pressure. Then for any position of the ball, as at B, Fig. 94, draw ordinate $BC = P$, the pressure on the spring. The straight line OCD will represent the uniform increase in spring pressure, as r increases. At some certain speed, the centrifugal force C of the ball at B will also $= BC$. Since C increases as r, it will equal P for any position of B at this speed and the governor is isochronous and unstable. At any other speed, either P or C will be greater and the governor will be inoperative.

Now, assume the spring pressure to be zero when the ball is at some position, as M in Fig. 95, and assume P to increase faster

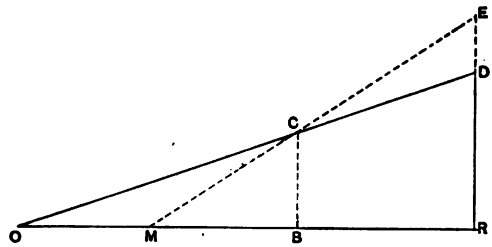


Fig. 95.

than C. Let OD in the figure show the increase in C and ME the increase in P. $P = C$ at the position B of the ball, and the governor is in equilibrium only at that point. Between M and B, C is greater and urges the ball out. Between B and R, P is greater and forces the ball back. This governor is stable.

Fig. 96 shows the effect of changing the speed of such a governor upon the position of the ball. The lines OD_1, OD_2, etc., mark the scales of C at different speeds, while the line ME gives the scale of the spring, as B_1, B_2, etc. The distances RD_1, RD_2, etc., are proportional to N^2.

Usually the spring tension is adjustable by means of a swivel nut at one end, and such adjustment would be shown in the figure by a change of the distance OM. A change of springs would be indicated by a new angle $E'MR$.

A change in the weight of ball would modify the slant of the C lines OD_1, etc. Such a figure enables the designer to predict the behavior of the governor under any conditions.

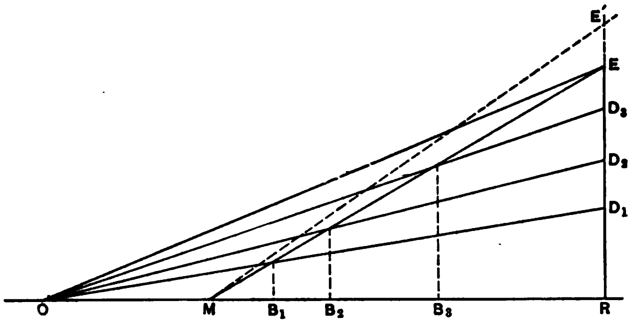


Fig. 96. Diagram Showing Effect of Speed Change.

87. Fluctuation of Shaft Governors.—In Fig. 97, let N be the position of ball for the slowest speed of the engine under maximum load, R the position for no load and B the position for normal speed under average load. ME is the scale of spring and OS, OC and OE the scales of centrifugal force at the different speeds.

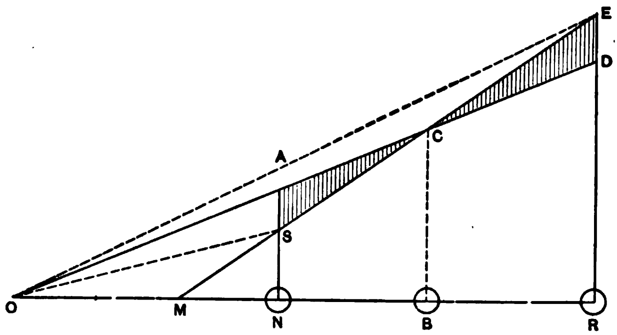


Fig. 97. Diagram Showing Fluctuation.

Suppose the engine to be running unloaded and the ball to be at R, both P and C being equal to RE. If the average load is suddenly applied, C is changed to RD, and there is an excess of spring pressure $= ED$. This forces the ball back to B, where

P and *C* again balance. But during this interval, kinetic energy has been stored in the ball, measured by the shaded area *EDC*.

If there were no friction and no further changes of speed, the ball would move on to *N*, overcoming the excess of *C* by its momentum. The centrifugal force in turn would send it back, and it would continue to oscillate back and forth indefinitely, like a pendulum. In practice, the friction of the moving parts will reduce the amount of oscillation and gradually bring the ball to rest at or near *C*. This is illustrated in Fig. 98, where *EF* repre-

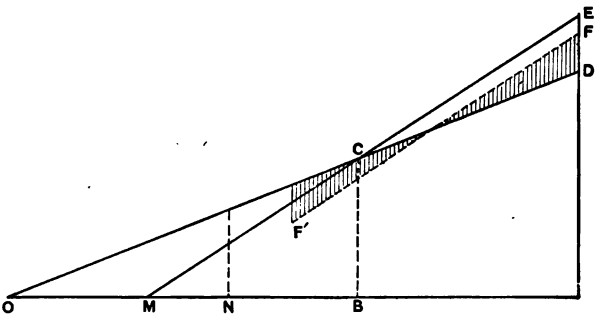


Fig. 98. Showing Effect of Friction.

sents the friction acting as a constant force to diminish *P*. The stored energy and, consequently, the amplitude of the vibration is reduced, and the ball stops short of *N*. The same effect is produced on the return, and the ball soon comes to rest at some point on one side of *C*.

A dashpot has a more favorable effect than ordinary sliding friction, since its resistance varies with the speed of oscillation, approaching zero as the ball comes to rest and allowing the latter to reach its true position of equilibrium at *C*. For this reason, dashpots are usually attached to the moving weights on shaft governors.

88. Hunting of Governors.—In the preceding article, it has been assumed that the speed of the governor remains constant after the change of load has been effected, and that *OCD* is the scale line of the centrifugal force.

In Fig. 97, suppose that the engine is running under full load

with the governor ball at N and that a part of the load is suddenly removed, carrying the point of equilibrium to B by increase of speed. As the ball oscillates to and fro on either side of B, too much steam will be admitted between B and N and too little between B and R. This in itself will cause a change of speed and a shifting of the point C.

When the ball is to the left of B, the speed will be increasing and C will move to the right. The reverse is the case when the ball is to the right of B. In common parlance, the governor is "hunting" for the point of equilibrium. The more sensitive the regulation, the more pronounced is this defect.

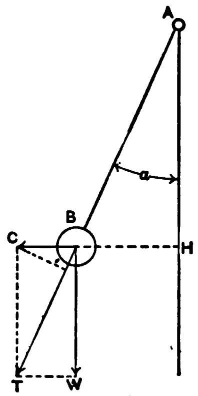


Fig. 99.

The same peculiarity exists in the pendulum governor. If the balls are in an extreme position, either up or down, and there is a sudden change of load, inertia will carry the balls by the position of equilibrium and cause oscillation. The alternate opening and closing of the steam valve due to this motion causes a variation of speed and consequent "hunting." A heavy fly wheel, by its inertia, tends to discourage hunting, but the best remedy is a dashpot which will absorb the fluctuation of energy and bring the governor promptly to equilibrium.

89. Oscillation of Pendulum Governors.—A diagram similar to Fig. 97 may be drawn for the pendulum governor, but moments must be used instead of forces. If, in Fig. 99, the

length of arm $AB = l$ and the angle $BAH = a$, the turning moments about A will be:

$$\text{Downward } Wr = Wl \sin a$$
$$\text{Upward } Ch = Cl \cos a$$
$$\text{But } C = \frac{WN^2 r}{35{,}200} \qquad (19)$$

and therefore, when N is constant, C is proportional to r

$$\text{or} \qquad C = Kr = Kl \sin a$$
$$\text{and} \qquad Ch = Kl^2 \sin a \cos a$$
$$= Kl^2 \times \frac{1}{2} \sin 2a$$

The upward turning moment is, therefore, proportional to sin $2a$ and the downward to sin a. In Fig. 100, the horizontal line, represents the value of angle BAH or a measured in degrees of arc. The sine curve marked W' has values of sin a for its ordinates corresponding to the downward moment produced by W.

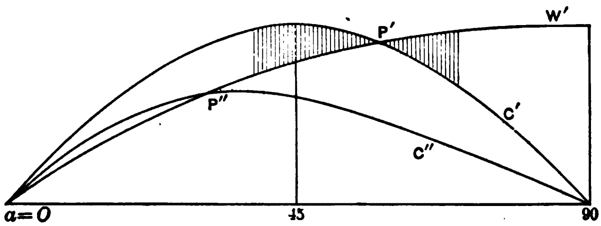


Fig. 100. Diagram of Governor Action.

The sine curve marked C' represents the upward moment due to C, its ordinates being proportional to sin $2a$. The two curves intersect at P', the point of equilibrium for this particular speed, when

$$W \sin a = C \cos a \qquad \text{(a)}$$

At some lower speed, C'' would be the curve of C, and P'' the point of equilibrium. The shaded areas represent fluctuation, the same as in Fig. 97, and in fact, the diagram in Fig. 100 may be studied in just the same manner as those used for spring gov-

ernors. A study of the curves in Fig. 100 shows that a given change of speed affects the position of equilibrium more at the left of the figure when a is small; i.e. the governor is more sensitive when the balls are down than when they are up. This has already been proved in Art. 83.

90. Lever Shaft Governors.—The simple type of governor shown in Fig. 93 is hardly practicable from a mechanical standpoint, and the construction shown in Fig. 101 is commonly substituted for it. The weight B is attached to a lever turning about

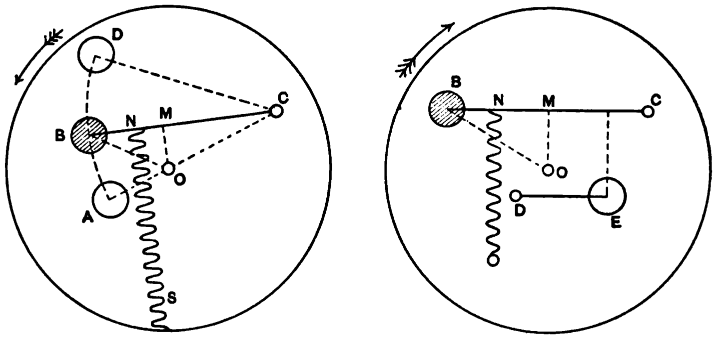


Fig. 101. Fig. 102.
Types of Shaft Governors.

C and moves in the arc of a circle. The spring is attached to some intermediate point, as N, and is normal to the lever when the latter is in its normal position. For sensitiveness the spring should have no tension when the lever is at CA, since there will be no component of C tending to move the lever out from that position.

In position CB, C will be proportional to OB, and the turning component proportional to OM. Calling this component C^1 and the pull of the spring P, the equation for equilibrium is:

$$C^1 \times BC = P \times NC$$

if the weight of the lever is neglected.

Isochronism cannot be attained by this device, since OM and SN do not remain parallel. In practice, the spring tension in-

creases faster than C^1, and the zero for the spring is above CA. The diagram would be somewhat similar to that in Fig. 97.

91. Shaft Governor with Eccentric.—The usual purpose of a shaft governor is to control a shifting eccentric. See Art. 56.

In Fig. 102, let E represent the mass of the eccentric and let it be pivoted to swing about D. The centrifugal moment of E is then opposed to that of B, and it decreases as the latter increases, since the eccentric is necessarily moved nearer the shaft center as the speed increases.

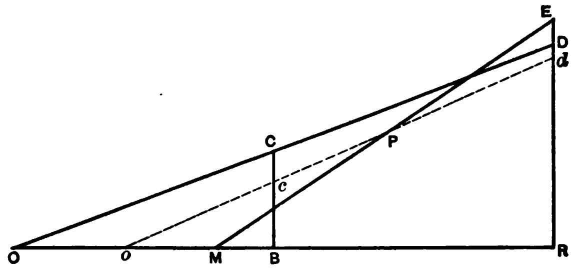


Fig. 103. Diagram of Moments.

Let OCD in Fig. 103 be the scale line of centrifugal moment in the governor without the eccentric and let Cc and Dd be the moments of the eccentric for the two extreme positions of the governor. (All moments are referred to the one center C in Fig. 102.) The dotted line ocd would be the scale of resultant moments, and it is this line which must be compared with the line of spring moments ME; i.e. in the figure, P would be the position of equilibrium. Neither CD nor cd would be exact straight lines.

92. Inertia Governors.—The inertia of a revolving weight instead of its centrifugal force is frequently used to move a regulator, or the two may be used in conjunction. If the governor shown in Fig. 101 turns in the direction indicated by the arrow, the inertia of the weight B will help the centrifugal force and make the governor more prompt in its action. If the engine is running at normal speed and the load is suddenly removed, the governor wheel will be accelerated, but the inertia of B will hold it back, and it will move towards the relative position D. In a

similar manner, if the load is suddenly increased, the wheel will be retarded while the momentum of B will cause it to surge ahead towards position A.

In the first instance, inertia helps centrifugal force in checking the speed, and in the second, inertia helps spring tension to open the valve promptly.

If the governor turns in the opposite direction, as indicated by the arrow in Fig. 102, the inertia of the weight retards the action of the governor and makes it more sluggish. Fig. 104 illustrates the Rites governor, which is probably in more general use than any other type of inertia governor. It consists of a rigid frame pivoted at some point C, carrying two weights B and b, and controlled by a spring which is attached at N.

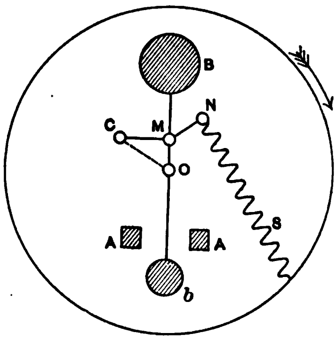


Fig. 104. Diagram of Rites Governor.

The relation between the weights is so adjusted that the center of gravity of the whole frame is at M, so that the resultant centrifugal force will act nearly at right angles to CM. The spring S is attached in such a way that the line of spring SN shall be nearly at right angles to the line joining N with C.

An increase of speed in the engine will cause M to move outward from O, and the weight B will swing to the left, practically shutting off steam. Disregarding inertia, this is the same as the governor in Fig. 102 would be with the weight concentrated at M.

If the governor is turning, as shown by the arrow in Fig. 104,

a sudden increase in speed of the wheel will cause the weights to lag behind, and therefore, to swing B to the left as before. The inertia of the whole mass is thus available to help the centrifugal force in checking the speed.

By making B and b nearly equal, thus bringing M near the center O, a governor is secured which has a relatively small centrifugal force and a light spring, while at the same time it has a large inertia effect, and is powerful. The connection of the spring at N is slotted so as to make it possible to change the distance from C to N. The tension of the spring is also adjustable, and the weight at B and b can be changed. This makes possible an adjustment of the governor for varying conditions and for any desired degree of sensitiveness.

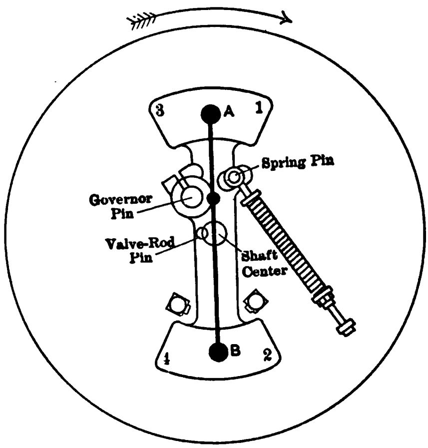


Fig. 105. Rites Governor.

93. Shifting Eccentrics.—Any governor may regulate the steam supply in one of two ways: (a) By operating on a throttle valve. (b) By changing the point of cut-off of the valve in the steam chest. The first method is not as a rule favorable to close regulation, since the throttle valve is located at some distance from the cylinder and a comparatively large clearance volume between the valve and the piston retards the action upon the latter.

Those governors which operate to vary the point of cut-off usually act upon either a shifting eccentric, some form of link motion, or a releasing device, as in the Corliss gear. The shifting eccentric is generally the accompaniment of the shaft governor. In the Rites governor, this consists of a crank pin or an eccentric attached directly to the frame which carries the weights and moving with it. See Fig. 105. The movement of *A* to the left carries the center of eccentric nearer the shaft center, diminishing the eccentricity and increasing the angular advance. See Fig. 52.

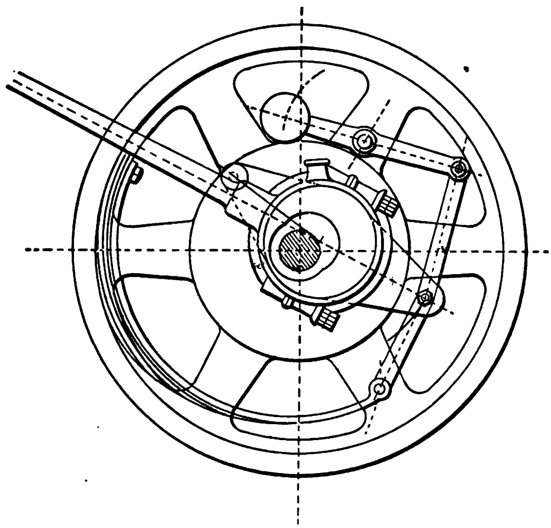


Fig. 106. Sweet Governor.

The simplicity of this governor, its almost perfect balance and its freedom from friction are strong points in its favor.

Centrifugal shaft governors usually operate an eccentric similar to that shown in Fig. 50. Links connect the weight levers to the arm carrying the eccentric and move it in or out as the speed varies. Fig. 106 shows a simple example of this class, the Sweet governor, having but one weight and employing a leaf spring, instead of the usual helical form.

94. The Allen Link.—The Porter or loaded governor (see Fig. 86) is often used in connection with a peculiar link motion known as the Fink or Allen gear. See Fig. 107. This gear consists of an eccentric having its center on the center line of crank and an eccentricity equal to the steam lap plus the lead. The link forms a part of the eccentric strap, and is pivoted at the lower end to a vertical hanger. The motion of the upper end of the link partakes of the horizontal motion of the hanger, and also has a tipping motion due to the vertical throw of the eccentric. A sliding block carried by a radius rod is moved up and down in the link by the action of the governor. The radius rod drives the steam valve through the medium of a rocker and valve

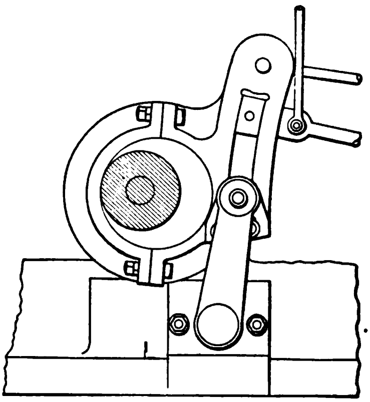


Fig. 107. Allen Link.

stem. The exhaust valve is driven in a similar manner by a connecting rod attached to the upper end of the link. Fig. 108 gives a skeleton diagram of the motion in which C is the crank pin, E the center of the eccentric, ELQ the link and HL the hanger. PR is the radius rod which drives the valve and the point P moves up and down on LQ.

If PR moves the valve directly without a rocker, E should be opposite C, as in the figure. The motion of P is then compounded of the motion of L and the tipping motion of the link about L. The former motion is that due to an eccentric OE with 90 deg.

angular advance. The latter motion is due to the vertical motion of the point E multiplied by the ratio $\frac{PL}{EL}$ of the two arms of the link. This is equivalent to the motion produced by an eccentric having no angular advance and a radius OP, Fig. 109, $= OE \times \frac{PL}{EL}$.

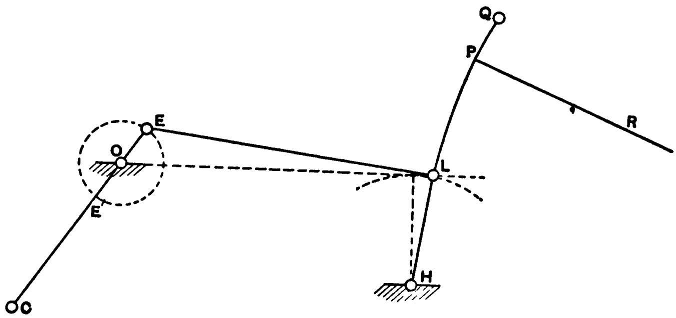


Fig. 108. Diagram of Allen Link Motion.

Combining OE and OP in Fig. 109, we have OM as the radius of the equivalent eccentric. In a similar manner for the point Q in Fig. 108, the equivalent eccentric ON is found in Fig. 109.

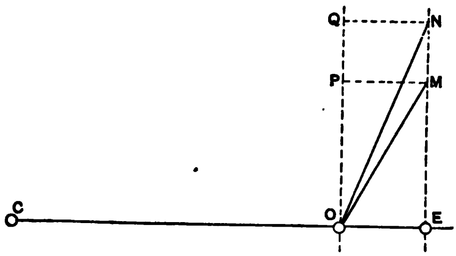


Fig. 109.

The Allen link motion is thus shown to be the equivalent of a shifting eccentric with a constant lead similar to that shown in Figs. 49 and 51. There are some modifications in practice which need to be noticed.

By raising or lowering the point H, the equality of lead and port openings in the two strokes is regulated. The makers of the Porter-Allen engine say that this point should be so adjusted that the arc of motion of L shall be tangent to the center line of engine. Furthermore, the motion of L is always in error on account of the obliquity of EL. To neutralize this, the makers use a rocker to reverse the motion of the valve and put E on the line OC at E'. The ratio of EL to OE is made the same as the ratio of connecting rod to crank in the main connection and thus one error offsets the other, so that the lead and cut-off can both be equalized.

95. The Corliss Valve Gear.—When a Corliss valve motion is used, the function of the governor is merely to release the valve which is then closed by the action of a spring or dashpot. The valve in this case is a segment of a cylinder, and has an oscillating motion about its axis. In Fig. 110, S is a cross-section of the

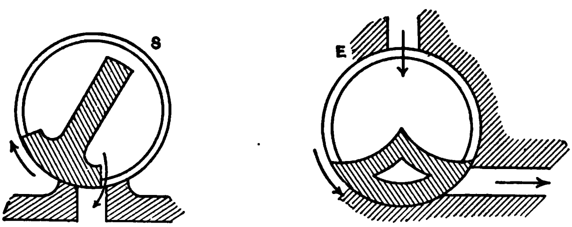


Fig. 110. Steam and Exhaust Valves of Corliss Engine.

steam valve just opening at the beginning of the stroke, while E shows the corresponding position of the exhaust valve. The latter is actuated by a crank and link motion, so as to move rapidly when opening or closing and slowly when wide open or closed. The steam valve is opened by a similar mechanism, is then released from its connections by the governor and closed suddenly by a dashpot. The releasing mechanism consists of a latch or hook opened at the proper time by means of a cam whose position is controlled by the governor. The advantage of this valve gear is the promptness of closing and the consequent maintenance of full steam pressure up to the time of cut-off.

The disadvantage consists in the necessity for picking up and releasing the valve, which prevents successful operation at high speeds. One hundred revolutions per minute is about the practical limit of speed for such gears.

96. The Joy Valve Gear.—The mechanism of the Joy valve gear is actuated directly by the connecting rod of the engine without the introduction of any eccentric. This motion is well adapted to vertical engines of the marine type, where the space on the crank shaft is too limited to allow of the presence of eccentrics.

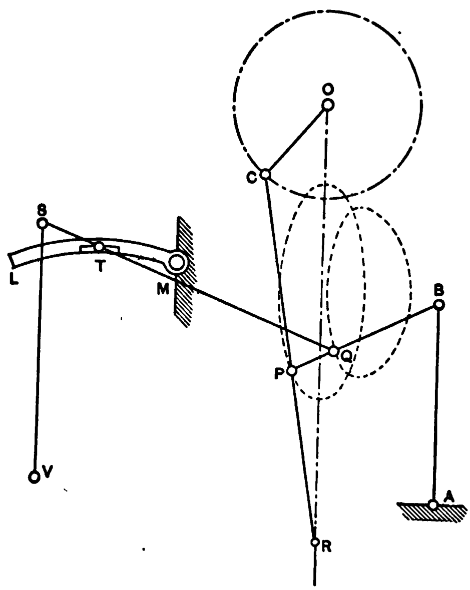


Fig. 111. Diagram of Joy Valve Gear.

In Fig. 111, let CR represent the connecting rod of the engine and P some intermediate point on the rod. The link PB is attached to P at one end and to a bridle lever BA at the other Some point Q on PB will move in an irregular curve, as shown by the dotted line through Q.

The valve lever STQ is actuated at one end by the point Q, while the fulcrum T is attached to a block sliding freely in the

adjustable link LM. The other end of the lever S moves the valve by means of the radius rod SV. The link LM is curved to a radius $= SV$, and is pivoted at some point M. It may be controlled by hand or by the action of a governor. When LM is horizontal, the point V receives only a reduced copy of the vertical motion of Q, since the horizontal components are absorbed by the sliding of T in the slot. This motion of V is nearly the same as would be produced by an eccentric opposite the crank, with an angular advance of 90 deg. When the link is inclined, V receives more or less of the motion due to the sliding of T in the slot; that is, the horizontal motion of Q. This is nearly equivalent to the motion produced by an eccentric at right angles to the crank.

The resultant motion is similar to that shown in Fig. 109, that of a virtual eccentric OM or ON, whose radius and angular advance depends upon the magnitude of the secondary motion due to the tipping of the link.

The true motion of V may be better understood by considering that the path of S is an ellipse whose major axis is more or less inclined to the horizontal according to the position of the link LM. The magnitude and the character of the motion of both S and V depend upon the location of the points P and Q upon their respective levers and can best be determined by accurate construction.

97. Safety Governors.—The fact has already been remarked that any pendulum governor connected to the engine shaft by belts or gears is an unsafe device, since a failure of the connection would result in dropping of the governor balls, opening of the steam valve and destructive racing of the engine.

In Corliss valve gears this danger is obviated by a permanent release of the latch mechanism when the governor falls below a certain point. The valve is not picked up by the latch, steam cannot get into the cylinder, and the engine stops. On large engines, a second governor is sometimes provided, which revolves continuously with the engine, but which does no work at safe speeds. When from any cause the speed increases beyond a certain limit, this governor releases a weight or powerful spring by means of a trip mechanism. The weight or spring thus released closes a throttle valve in the steam pipe and stops the engine.

PROBLEMS.

1. Determine the height of cone in inches for a simple pendulum governor when running 75 revolutions and when running 250 revolutions per minute.

2. Determine the height of cone of a Porter loaded governor similar to that shown in Fig. 86, at 250 revolutions per minute, if the balls weigh 1 lb. each and the center load is 30 lb.

3. What would be the height of cone of the governor in Fig. 87 under similar conditions?

4. Three governors have the following dimensions:

$$\begin{aligned} &\text{Height of cone} &&= 8.66 \text{ in.} \\ &\text{Radius of revolution} &&= 5 \quad \text{in.} \end{aligned}$$

(the angle of inclination being 30 deg. with the vertical). The governors are of the shape shown in Fig. 88, and the cross arm C is distant vertically 3 in. from the vertex A. Determine the speed of revolution, and the change in height of cone in each governor for a fluctuation of 5 per cent in the speed.

5. A plain shaft governor, Fig. 93, has a weight of 54 lb. moving radially from $r = 18$ in. to $r = 22$ in. The spring is at zero stress when $r = 6$ in. Determine the values of the centrifugal force and of the spring stress when $r = 18$ in., 20 in. and 22 in. at 175 revolutions per minute.

Find also the scale of spring and the least and greatest speeds of the governor.

CHAPTER IX.

FLY WHEELS.

98. The Function of a Fly Wheel.—It has been shown in the preceding chapter that the governor of an engine controls the average speed within certain limits, by regulating the steam supply to correspond to the work to be done.

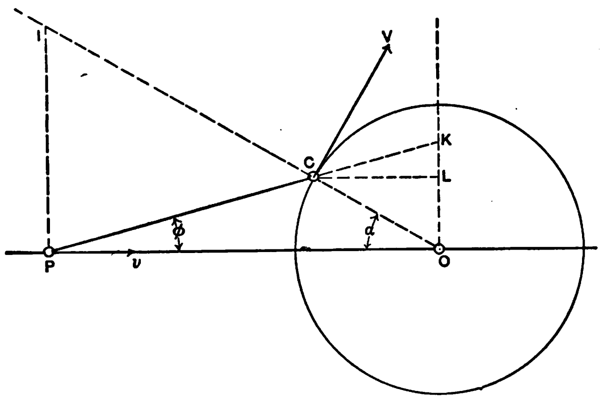


Fig. 112. Diagram of Relative Velocities of Piston and Crank Pin.

There are certain minor fluctuations of speed over which the governor has no control, and it is the function of the fly wheel to control these and prevent them from becoming excessive. These changes of speed occur during each revolution and are due to variations in the tangential pressure on the crank pin. The primary causes of these variations in pressure are as follows:

(a) Variation in the steam pressure on the piston, due to admission, expansion, release and compression.

(b) The inertia forces of the piston, crosshead and connecting rod, arising from their reciprocating motion.

(c) The nature of the crank and connecting-rod motion, which is such as to vary the useful pressure on the crank pin from zero to a maximum.

Of these, the last is by far the most important, and would necessitate the employment of a fly wheel, were the other sources of variation absent.

99. The Velocity of the Piston.—In order to study the balance of forces in the engine mechanism, it is necessary to determine the relative velocities of the various parts.

In Fig. 112, let P be the crosshead of an engine, PC be the connecting rod and OC be the crank. Let V be the velocity of the crank pin and v the velocity of the piston and crosshead at any instant. Erect a perpendicular on OP at P and produce OC to meet the perpendicular at I.

Then will I be the instant center about which PC is turning and the velocity ratio of P and C will be:

$$\frac{v}{V}=\frac{IP}{IC}$$

Erect a perpendicular on OP at O and produce PC to meet this perpendicular at K. Then will the triangles IPC and OKC be similar; therefore,

$$\frac{v}{V}=\frac{IP}{IC}=\frac{OK}{OC} \tag{25}$$

If V is taken to such a scale that $V=OC$, then will $v=OK$. Draw the horizontal line CL; the angle $KCL =$ angle CPO.

As the connecting rod becomes longer, these angles diminish and the point K approaches L. OL is then the velocity due to the crank and KL an additional velocity due to the angularity of the rod. When C is to the right of the center O, KL becomes negative, and the velocity is less than that due to the crank alone.

100. Useful Effect on the Pin.—Since the work done on the crank pin is the same as that done on the piston, the pressure ratio must be the reciprocal of the velocity ratio. Therefore, if p is the pressure on the piston and P the tangential pressure on the pin:

$$\frac{P}{p}=\frac{v}{V}=\frac{OK}{OC} \tag{26}$$

In Fig. 113, let $HIMNR$ be the diagram of pressures on the piston, then will PQ be the pressure for any position P of the

crosshead. On the corresponding crank line CO lay off $Co = PQ$ and draw the vertical ok, and by equation (26) ok will represent the tangential pressure on C. For,

$$\frac{ok}{oC} = \frac{OK}{OC} = \frac{P}{p}$$

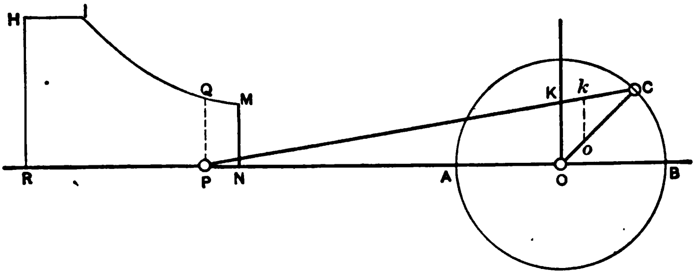


Fig. 113. Diagram for Tangential Pressures.

In Fig. 114 lay off the horizontal line to represent the circumference of the crank circle and divide it into a suitable number of equal parts. Let the point o represent the position of C as in Fig. 113. Draw the ordinate ok equal to ok in Fig. 113. A

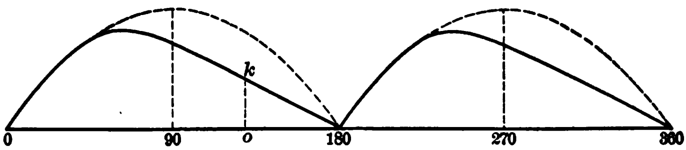


Fig. 114. Curves of Tangential Pressures.

series of ordinates so erected will determine the curve of tangential pressure on the crank pin, as shown by the full lines. The area under each curve will represent the work done on the pin during one stroke and will equal the area of the corresponding indicator diagram.

The dotted curves in Fig. 114 are sinusoids and show the effect on the pin when the piston pressure is uniform and the connecting rod has no angularity.

101. Inertia of the Reciprocating Parts.—As a matter of

fact, not all of the pressure upon the piston reaches the crank pin on time. During the first part of the stroke, the mass of the reciprocating parts is gradually accelerated from rest to its greatest velocity near mid-stroke and a part of the steam pressure is used in this way.

During the latter part of the stroke, the mass is similarly retarded by the reaction of the crank pin, and the kinetic energy stored during the acceleration is now restored as work on the pin.

Referring again to Fig. 112, denote the angle COP by α and the angle CPO by ϕ. Let V be drawn to such a scale as to equal OC.

Then $v = OK = OL \pm LK$

$$= V \sin \alpha \pm CK \sin \phi \qquad \text{(a)}$$

As ϕ is usually a small angle, approximate by taking $CK = CL = V \cos \alpha$

Then $LK \doteqdot V \cos \alpha \sin \phi$, nearly (b)

Let $OC = r$ and $PC = l = nr$.

In the triangle POC

$$\sin \phi : \sin \alpha = r : nr$$

and
$$\sin \phi = \frac{\sin \alpha}{n}$$

Substituting this value of $\sin \phi$ in (b):

$$LK = \frac{V \cos \alpha \sin \alpha}{n} \qquad \text{(c)}$$

and finally substituting in (a)

$$v = V \sin \alpha \pm \frac{V \cos \alpha \sin \alpha}{n}$$

$$= V \sin \alpha \left(1 \pm \frac{\cos \alpha}{n}\right) \qquad (27)$$

To determine the acceleration, it is necessary to differentiate this value of v with respect to the time t.

In the time dt, let dv, $d\alpha$, etc., be the various differentials of v, α, etc.

Then the space passed over by C in the time dt is:

$$r\,d\alpha = V\,dt$$

and
$$\frac{d\alpha}{dt} = \frac{V}{r} \qquad \text{(d)}$$

Differentiating (27)

$$\frac{dv}{d\alpha} = V \cos \alpha \pm \frac{V}{n} (\cos^2 \alpha - \sin^2 \alpha)$$

$$= V \cos \alpha \pm \frac{V}{n} \cos 2 \alpha \qquad \text{(e)}$$

Multiplying (e) by (d)

$$\frac{dv}{dt} = \frac{V^2 \cos \alpha}{r} \pm \frac{V^2}{nr} \cos 2 \alpha \qquad (28)$$

This is the rate of acceleration of the reciprocating parts, and as the accelerating force F is equal to the mass $\times$ the accelara-tion:

$$F = \frac{WV^2 \cos \alpha}{gr} \pm \frac{WV^2}{ngr} \cos 2 \alpha \qquad (29)$$

where $\frac{W}{g}$ is the mass of the reciprocating parts.

(29) may be expressed in the simpler form:

$$F = \frac{WV^2}{gr} \left(\cos \alpha \pm \frac{\cos^2 2 \alpha}{n} \right) \qquad (30)$$

The curve representing these two equations may be best drawn by plotting the first part separately and then combining the second with it.

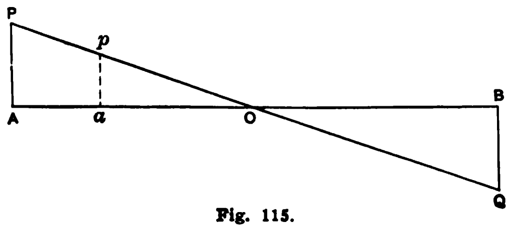


Fig. 115.

In Fig. 115 let AB represent the stroke of the piston and let AP and BQ each equal $\frac{WV^2}{gr}$. Then will the straight line POQ represent $\frac{WV^2 \cos \alpha}{gr}$ for various values of α.

This is the part of the accelerating force due to the crank motion when the angularity of the rod is disregarded. It is the same as the horizontal component of the centrifugal force would be if a

weight equal to that of the reciprocating parts were concentrated at the crank pin and revolving with it. A graphic method of expressing the remainder of the equation is given by Professor Ripper in his book on the Steam Engine and credited by him to J. W. Kershaw.

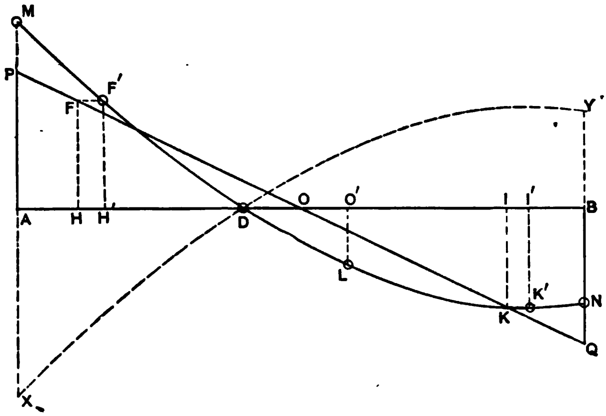


Fig. 116. Diagram of Accelerating and Retarding Forces due to Inertia.

In Fig. 116, let AB represent the stroke of piston and let AP and BQ as before represent $\frac{WV^2}{gr}$ · Let H, O and I be the positions of pistons corresponding to crank angles of 45 deg., 90 deg. and 135 deg. when angularity of rod is disregarded, and let H^1, O^1 and I^1 be the true positions of piston for those angles. The values of the expression $\frac{WV^2}{ngr}\cos 2\alpha$ for different values of α are as follows:

For $\alpha = 0$	the	expression	becomes,	$\frac{WV^2}{ngr}$
" " "	45 deg.	"	"	0
" " "	90 "	"	"	$-\frac{WV^2}{ngr}$
" " "	135 "	"	"	0
" " "	180 "	"	"	$\frac{WV^2}{ngr}$

In Fig. 116, therefore, make PM, $O'L$ and QN each $\frac{1}{n}$ of AP. Draw the perpendiculars HF and IK to give the values of $\frac{WV^2}{gr}\cos\alpha$ for 45 and 135 deg. As the value of $\frac{WV^2}{ngr}\cos 2\alpha$ is zero for both these positions, make $H'F' = HF$ and $I'K' = IK$. Locate the point D by finding the position of the piston, when the crank and connecting rod are at right angles. This is approximately the position where v is a maximum, and therefore, the acceleration is zero. (This may be seen from Fig. 112 by considering that OK will be a maximum when PC is tangent to the crank circle.)

Draw a free curve through the points $MF'DLK'N$ and its ordinates will represent the accelerating and retarding forces due to the inertia of the reciprocating parts for any position of the piston.

The area $MF'DA$ shows the energy stored during the acceleration and the area $DLKNB$ shows the energy restored to the crank pin during the retardation. If the figure is correctly drawn, these two areas will be equal. The dotted line YDX shows the variation of the inertia force during the return stroke. It must be remembered that A is the head and B the crank end of the cylinder in the figure.

102. Net Pressures.—In order to draw a diagram which shall correctly represent the tangential forces acting on the crank pin, it will be necessary to begin with the actual indicator diagrams, to determine the net pressure on the piston, to make allowance for the inertia forces and finally, by the methods of Art. 100, to draw the crank effort diagram.

Let H and C in Fig. 117 be the head and crank indicator diagrams, respectively. Divide the base line into equal parts and erect ordinates at the division points.

In Figs. 118 and 119 lay off AB, divide and erect ordinates as before. On each ordinate in Fig. 118 lay off a distance equal to the difference between the forward pressure of the H diagram and the back pressure of the C diagram in Fig. 117; e.g. PQ in Fig. 118 $= PQ$ in Fig. 117. The line $RPST$ joining the tops of these ordinates shows the variation in the net pressure on the

piston during the forward stroke. A similar curve for the return stroke is found as in Fig. 119.

To allow for the influence of the inertia forces, the inertia curves MDN are drawn to scale on each diagram, the ordinates

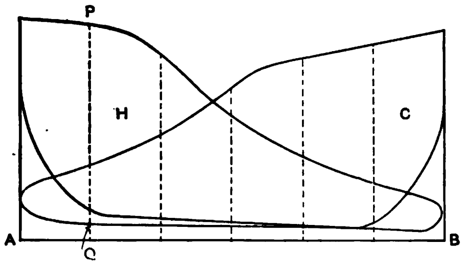


Fig. 117. Pressures as Shown by Indicator Diagrams.

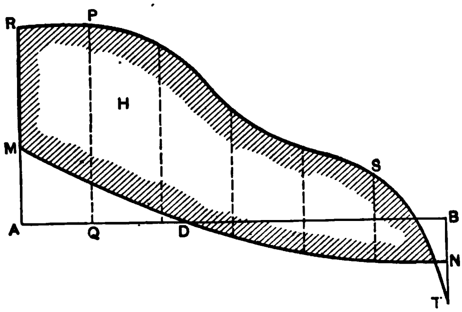


Fig. 118. Horizontal Forces on Crank Pin, Forward Stroke.

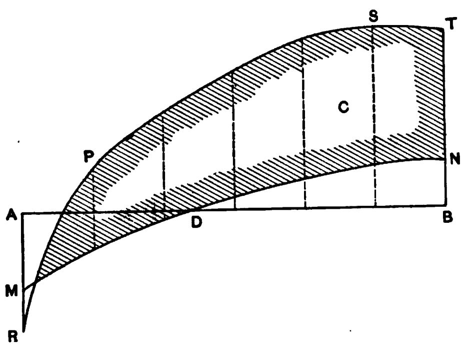


Fig. 119. Horizontal Forces, Return Stroke.

of these curves representing the values of the accelerating force in pounds per square inch of piston area. For instance, AM is equal to

$$\frac{WV^2}{gr}\left(1+\frac{1}{n}\right)$$

divided by area of piston. (See Fig. 116.) The ordinates of the shaded diagrams in Figs. 118 and 119 thus represent to scale the net forces acting horizontally to turn the crank pin. The curves of tangential pressure may be drawn as in Figs. 113 and 114 by substituting these revised diagrams for the indicator diagram.

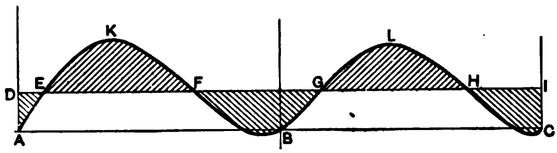


Fig. 120. Tangential Forces.

Fig. 120 shows such a curve $AKBLC$ drawn from the diagrams in Figs. 118 and 119. In this figure, the base line ABC is equal to the circumference of the crank pin circle, and any ordinate of the curve represents to scale the tangential pressure on the crank pin at that instant. If drawn to the same scale as the diagrams in the two preceding figures, the areas will be the same, since the work done on the crank pin is equal to that done on the piston.

103. Fluctuation of Energy.—If the area of the diagram in Fig. 120 is divided by the length AC, the quotient will be the mean pressure. Lay off AD and CI equal to this quotient and draw the line of mean pressure DI. This may also be assumed to be the line of uniform resistance, referred to the crank pin. The areas EKF and GLH above this line represent excess of energy exerted during these intervals, while the areas DEA, FBG and HIC represent corresponding deficiencies. If there are heavy rotating pieces on the shaft, such as cranks, eccentrics and fly wheels, the velocity of these will be accelerated during the intervals EF and GH, and the excess of energy will be stored as kinetic energy of rotation.

A corresponding retardation during the intervals DE, FG and

HI will cause the stored energy to be given up and converted into useful work. During the intervals from E to F and from G to H, the speed of the wheel will be increasing, so that F and H are points of maximum and E and G points of minimum rotative speed.

The ratio of the excess of energy EKF to the total energy AKB in one stroke is usually denominated $\frac{\Delta E}{E}$ and depends upon the various modifications which have been noticed in the preceding diagrams, such as ratio of expansion, connecting-rod ratio and weight of reciprocating parts.

Rankine gives the following values of the ratio $\frac{\Delta E}{E}$ for different conditions, assuming the connecting-rod ratio to be five:

Cut off at	$\frac{1}{2}$	$\frac{1}{3}$	$\frac{1}{4}$	$\frac{1}{5}$	$\frac{1}{6}$	$\frac{1}{7}$	$\frac{1}{8}$
Condensing . . .	......	.326	.346	.356	.368	.378	.382
Non-Condensing . .	.320	.372	.418	.464	......	......	......

In designing an engine, it is better to determine the ratio of fluctuation from the diagrams.

104. Weight of Fly Wheel.—The whole mass of the fly wheel and the other rotating parts helps to absorb the excess of energy ΔE, but in designing fly wheels, it is customary to disregard the effect of masses other than that of the wheel rim, and to make that heavy enough to reduce the speed variation within the desired limits. This practice is admissible, since the error is on the safe side.

Let $W =$ weight of wheel rim.

$\rho =$ its radius of gyration.

$v_1 =$ its greatest velocity.

$v_2 =$ its least velocity.

$v =$ its mean velocity.

$\frac{1}{m} =$ coefficient of speed variation.

$\Delta E =$ stored energy.

Then
$$\triangle E = \frac{W(v_1^2 - v_2^2)}{2g} \tag{a}$$

since the first term represents the excess of energy to be taken care of, and the second term represents the increase of kinetic energy due to the speed change. To express this in terms of the mean speed, let:

$$v_1 + v_2 = 2v$$
$$v_1 - v_2 = \frac{v}{m}$$

Factoring (a)
$$\triangle E = \frac{W}{2g}(v_1 + v_2)(v_1 - v_2) \tag{b}$$

Substituting the values for the sum and difference of v_1 and v_2, and reducing:

$$\triangle E = \frac{Wv^2}{gm} \tag{31}$$

The value of m may be as low as 50 for ordinary mill engines, but engines used for electrical transmission should have m equal to 100 or 150. The velocity in feet per second at the extremity of the radius of gyration is represented by v. The correct value for ρ is $\frac{1}{4}\sqrt{(d^2 + d_1^2)}$ where d and d_1 are the inner and outer diameters, but the value $\frac{1}{4}(d + d_1)$ is accurate enough for ordinary calculations. For thin rims, such as are used for combined belt and fly wheels, the outer radius of rim may be substituted for ρ. In such case, let

N = number of revolutions per minute.
d = diameter of wheel in feet.

Then $v = \frac{\pi d N}{60}$

Substituting this value in (31) and solving for W:

$$W = 11750\frac{m \triangle E}{d^2 N^2} \tag{32}$$

Example.—Let it be required to find the weight of a rim of a fly wheel 18 ft. in diameter for a 600 h.p. engine making 80 r.p.m.

The engine is non-condensing and has a cut-off at 1/5 stroke. As the engine is belted to a generator, it is desired to have $m = 100$.

The energy developed in one stroke will be:

$$E = \frac{600 \times 33,000}{160} = 123,800 \text{ ft. lb.}$$

From the table in Art. 103, $\frac{\Delta E}{E} = 0.464$, and in the absence of accurate diagrams, this value may be used:

$$\Delta E = 0.464\ E = 57,400 \text{ ft. lb.}$$

Substituting values in (32) and reducing,

$$W = 32,515 \text{ lb.}$$

Calling the weight of cast iron 450 lb. per cu. ft., the volume of the rim will be:

$$\frac{32,515}{450} = 72.26 \text{ cu. ft.}$$

Calling the mean circumference of rim 56 ft., the area of cross section of rim becomes:

$$\frac{72.25}{56} = 1.29 \text{ sq. ft.}$$

The general subject of the design of fly wheels will be more fully treated in Chapter XIV.

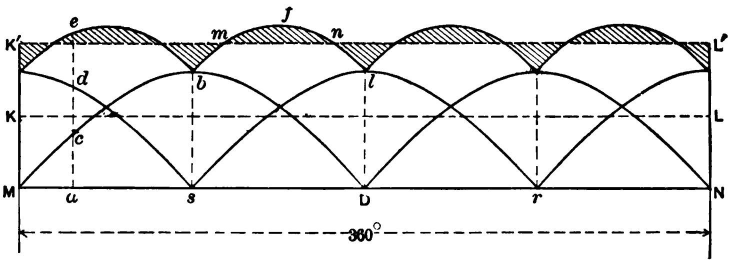


Fig. 121. Fluctuation of Energy, Two-crank Engine.

105. Two and Three Crank Engines.—The variation in turning effort and the consequent fluctuation of energy may be much reduced by using more than one crank on the shaft.

If two cranks are employed differing 90 deg. in phase, the

maximum turning effect of one will coincide in time with the minimum effect of the other, and the resultant effect will be quite uniform.

Fig. 121 illustrates the manner of combining the two diagrams: The two curves are superimposed with a relative displacement of 90 deg., as shown in the figure, and the ordinates of one curve are then added to the corresponding ordinates of the other to obtain points on the upper or resultant curve; e.g.,

$$ae = ac + ad.$$

The horizontal line of mean effort is then drawn, and the shaded areas show the excess and deficiency of energy. A comparison of this figure with that next preceding shows the great improvement in uniformity which has resulted. By using three cranks at angles of 120 deg., a still further improvement is effected.

The fluctuations of energy are now much less and occur at much shorter intervals. A fly wheel with a comparatively light rim will suffice, and in some kinds of service, the fly wheel may be omitted altogether.

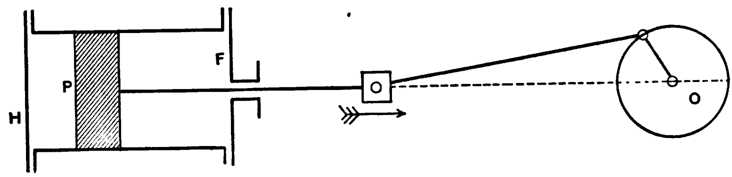


Fig. 122. Illustrating Engine Vibration.

106. Vibration of Engine.—In the preceding articles we have seen the effect of the inertia of the reciprocating parts upon the turning moment of the crank shaft. These same inertia forces have also a disturbing effect upon the stability of the engine frame. When the piston starts from the head end of the cylinder and moves forward towards the crank (Fig. 122), the steam pressure is the same on the head H and the piston P, tending to move H to the left and P to the right. If the motion of P were uniform, the pressure on P would be transmitted through the in-

tervening parts directly to the main journal O, and there would be a balance of right and left forces, producing a tension in the engine frame, but having no tendency to move the frame to right or left. As we have already seen, all of the pressure on P is not transmitted to O, but a part is absorbed in accelerating the mass of the reciprocating parts. Consequently, the pressure on O is less than that on H, and there is a tendency to move the engine frame to the left. During the latter part of the stroke, the momentum of the reciprocating parts will increase the pressure on O, making it greater than the steam pressure, and the engine will tend to move to the right. Similar effects are noticed on the return stroke. In general, the engine tends to move to the left when the piston is nearer the left end of the cylinder, and to the right when the piston is nearer the right end. The heavier the reciprocating parts and the greater the speed, the more noticeable the vibration.

Early compression or cushioning has been supposed by some to mitigate this evil, but this is not the case. Suppose that the piston is approaching the right end of the cylinder, that the steam pressure is 2500 lb. and the inertia force is 500 lb., the pressure on H will then be 2500 lb. and on O $2500 + 500 = 3000$ lb. There will be an unbalanced force of 500 lb. tending to push the engine frame to the right.

Suppose further that the back pressure due to compression is 250 lb. This is exerted equally on P and the head F. The pressure transmitted to O is now $2500 - 250 + 500 = 2750$ lb. There is now an unbalanced pressure of 250 lb. at O and another similar pressure at F, or a total of 500 lb. as before.

The only effect of the cushion is to transfer a part of the unbalanced pressure from O to F without changing the effect on the frame as a whole. In other words, cushioning relieves the shock on the crank and the main journal, but not that on the frame considered as a unit.

107. Counterbalancing.—The forces just described can be partially neutralized by introducing a centrifugal force at the crank which shall be opposed to the action of the reciprocating parts at or near the ends of the stroke. If a crank is constructed without a counterbalance, its resultant centrifugal force will be

in the direction OC, Fig. 123, and will tend to increase the vibration of the engine.

If the crank is heavily counterweighted on the side opposite the crank pin, the resultant centrifugal force will be opposite to the unbalanced force of the reciprocating parts, when near the ends of the stroke, and the vibration will be reduced, Fig. 124. When the crank is near the 90-deg. points, the counterbalance acts as a disturbing element and tends to cause vibration at right angles to the line of stroke. This fact limits the weight of the counterbalance.

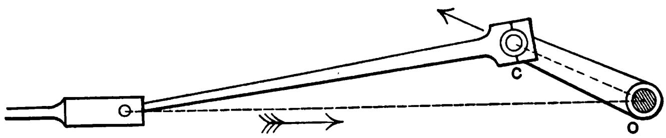


Fig. 123. Crank Without Counterbalance Weight

In determining the proper weight, it is customary to consider one third or one half of the connecting rod as a rotating piece and the remainder as a reciprocating piece. It is evident if enough counterweight is added to balance the crank itself and the rotating part of the connecting rod, that the resultant centrifugal force will be zero and the inertia of the reciprocating parts will be unbalanced.

This arrangement is more suitable for vertical engines where vibration in the line of stroke is less injurious than transverse vibration.

In a horizontal engine, running at a comparatively high speed, it is desirable that the counterweight be heavy enough to neutralize a portion at least of the inertia forces. This may be better understood by reference to Figs. 125 and 126. Let OC be any position of the crank and let Oc represent to scale the net centrifugal force of the crank and that part of the connecting rod which is assumed to be a rotating piece, as for instance, the part GC in Fig. 124. Let Oa and Ob represent to the same scale the inertia forces of the parts to the left of G, as determined by the diagram like Fig. 116; i.e. let $Oa = AM$ in that figure and $Ob = BN$. Take $am = bn = Oc$. Then will Om and On be the net

forces acting on the frame at the dead points. Let F and H be the two crank positions when the acceleration of the piston $= 0$, as at D in Fig. 116. Of and Oh each $= Oc$ will represent the forces acting acting at those times.

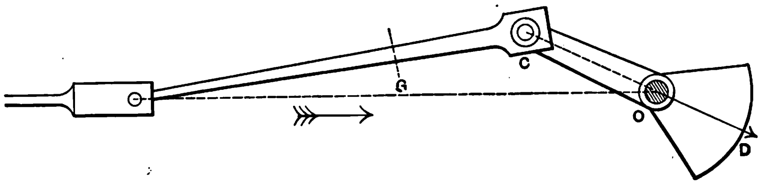


Fig. 124. Crank with Counterbalance Weight.

To find the resultant force for any crank position as OC, take $Od =$ the corresponding inertia force, as for instance $H'F'$ in Fig. 116. Combining this with the centrifugal force Oc gives the resultant force Oe acting on the frame at this time. Other points are found in like manner, and the resulting shaded curve $mhnf$ is drawn.

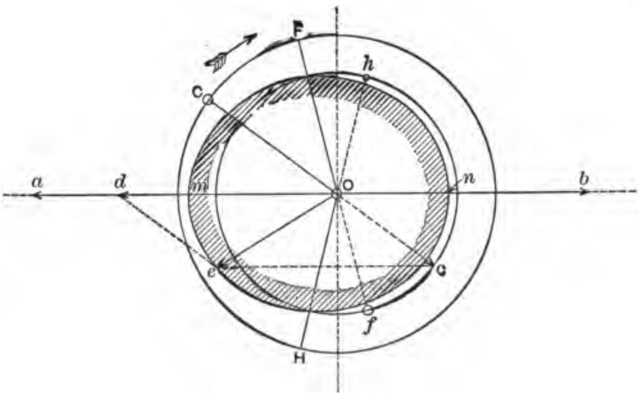


Fig. 125. Showing Effect of Counterbalancing.

In Fig. 125, the counterweight is sufficient to neutralize about one half of the inertia forces at the dead points, and the resulting diagram is approximately a circle; i.e. the vibration will be nearly the same in all directions. In Fig. 126, the counterweight is heavier and the curve of forces is kidney shaped with the longer

axis transverse. The vibration will then be greater at right angles to the line of stroke, an arrangement which is all right for horizontal engines, but unsuitable for those whose stroke is vertical.

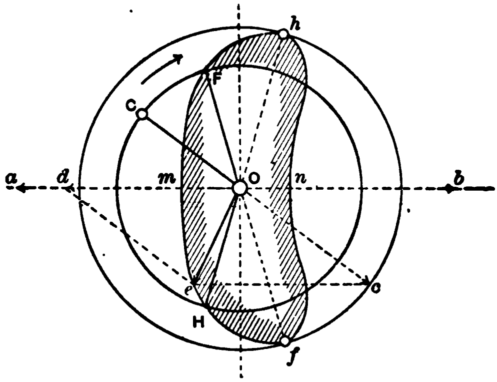


Fig. 126. Effect of Heavy Counterweight.

108. Unbalanced Cranks.—Fig. 127 shows the diagram of unbalanced forces for a crank which is insufficiently counterweighted. The net centrifugal force is now in the same direction

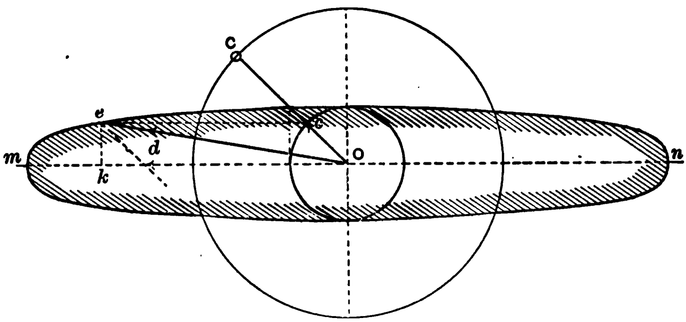


Fig. 127. Counterweight too Light.

as the crank (notice Oc in the figure), and its effect is to augment instead of neutralize the inertia forces. Such an arrangement might be tolerated in a vertical engine. The tendency to

longitudinal vibration may be shown more clearly by plotting a curve as in Fig. 128.

In this figure, the horizontal line *ABC* represents the circumference of the crank circle rectified. This is divided into equal

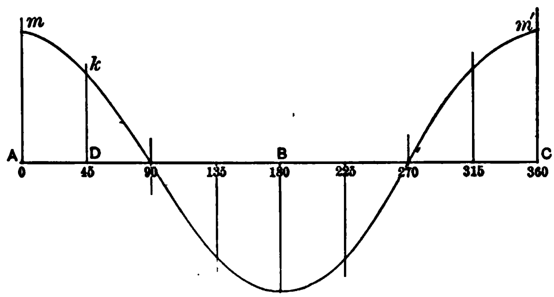


Fig. 128. Diagram of Longitudinal Forces of Fig. 127. (To a Reduced Scale.)

parts, and ordinates are erected at the points of division. On each ordinate is measured off the longitudinal component of the vibrating force for that instant. For instance, comparing this figure with the preceding:

$Am = Om.$
$Bn = On.$
$Dk = Ok,$ etc.

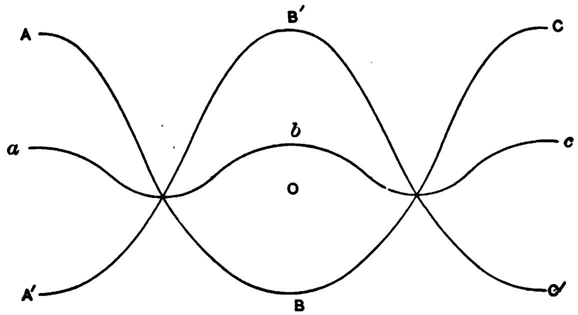


Fig. 129. Diagram for Two Cranks at 180 Degrees.

In case of two engines working in parallel with cranks at 180 deg. the diagram may be drawn as in Fig. 129. The two curves

ABC and *A'B'C'* are drawn separately for the two cranks and then combined in the resultant curve *abc* which gives the net effect on the frame.

If it were not for the angularity of the connecting rod, the resultant would be a straight line coinciding with the X axis, and the vibration would be nil.

In a similar manner, it may be shown that the effect of two cranks at 90 deg. is to increase the vibration, while three cranks at 120 deg. practically neutralize each other despite the angularity of the rod.

This latter fact may be demonstrated analytically by substitution in Equation (30), Art. 101. If three equations of this form containing the angles α, $\alpha + 120$ and $\alpha + 240$, respectively, be added together, member for member, the resulting equation will reduce to zero.

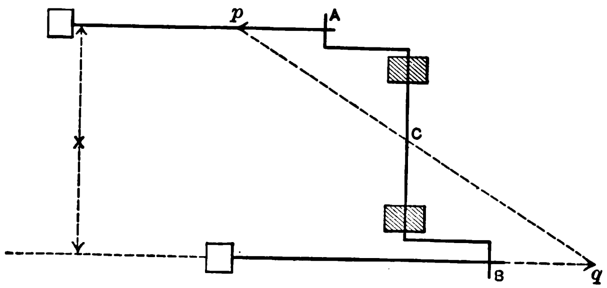


Fig. 130. Oscillating Effects, Cranks at 180 Degrees.

109. Oscillating Effects.—In the preceding article, the assumption has been made that the inertia forces of the several cylinders of an engine act at one point, and can be added together algebraically. This is manifestly impossible in practice, since the different cranks must rotate in different planes. Fig. 130 is a plan diagram of the cranks of an engine which are 180 deg. apart in rotation and which are the distance x apart in plan. The inertia forces acting at the instant are Ap for one crank and Bq for the other. These two form a couple, tending to turn the engine frame in a left-hand rotation. When the cranks shall have turned 180 deg., the direction of the couple will be reversed.

There is thus a tendency to oscillate the frame in a horizontal plane. This effect may be decidedly disagreeable in engines which have no secure foundations, as is the case with traction and marine engines.

It can be partially eliminated by counterbalancing or can be almost wholly overcome by a balancing of couples. If the cranks at *A* and *B* coincide in direction and a crank be inserted at *C* differing 180 deg. from the others, both forces and couples can be balanced by having the reciprocating parts of the C^1 engine as heavy as those of *A* and *B* combined. If uniformity of parts be desired, two cylinders can be used in the middle as shown in Fig. 131. It will be seen that *A* and *C* in the figure form a left-hand couple and *B* and *D* a right-hand one of the same moment.

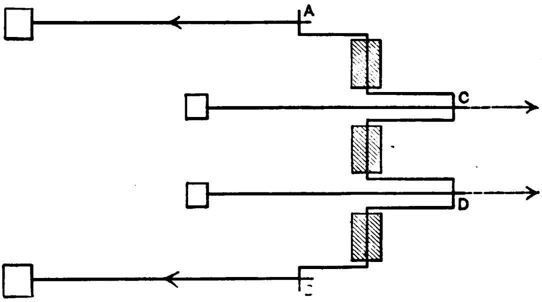


Fig. 131. Arrangement producing Balanced Couples.

110. Crank Models.—A good method of determining the amount of counterweight to be added to a crank and the balance of the crank as a whole, is to make a model of the crank and the end of the connecting rod to scale from a uniform quality of cardboard. One thickness of cardboard can be taken to represent a unit of thickness of the metal, as an inch or half an inch, and the different parts can be built up of the proper number of layers. The model can be balanced to determine the center of gravity, and the weight can be either calculated or determined by weighing against squares and rectangles of the same cardboard. This method is more expeditious than mathematical computations and usually more accurate.

Problems.

1. A crank has a radius of 9 in. and makes 180 r.p.m. The connecting rod is 45 in. long. Find the velocity of the piston when $\alpha = 30, 45, 60$, and 120 deg. Find its maximum velocity.

2. In the above engine, if the diameter of the piston is 14 in., what will be the tangential effort on the crank pin when $\alpha = 135$ deg., assuming the steam pressure at that time to be 20 lb.?

3. In the above engine, assuming the weight of the reciprocating parts to be 600 lb., find the value of the inertia force per sq. in. of piston when $\alpha = 0, 45, 135$, and 180 deg.

4. In problem 3, assuming the initial steam pressure to be 90 lb. gage and the back pressure 2 lb. gage, with a cut-off at $\frac{1}{5}$ stroke and no compression, find the net forward pressures for each 30 deg. of crank motion from zero to 360 deg.

5. In the preceding problem, find the net tangential effort on the crank pin for each 30 deg. of motion from zero to 360 deg. and draw a diagram.

6. Determine approximately in Problem 5 the ratio ΔE to E for one stroke.

7. Determine the weight of rim for a fly wheel 8 ft. in diameter to reduce the fluctuation of speed to one per cent, using data in preceding problems.

8. The following are the weights of reciprocating parts in the engine above described:

Piston and rod,	300 lb.
Crosshead,	180 "
$\frac{1}{2}$ Connecting rod,	120 "

Design a crank to counterbalance about one half the above weights and make a cardboard model of same one quarter size.

9. Draw to scale diagram of vibrating forces for the engine in Problem 8, similar to Fig. 125.

CHAPTER X.

STEAM IN THE CYLINDER.

111. Condensation.—One of the inconveniences attendant upon the use of steam in an engine is the presence of water in greater or less quantities. Not only is the presence of water prejudicial from an economical standpoint, but it is a source of danger in the cylinder. It is especially necessary to guard against the effects of water in first starting an engine, when all the exposed surfaces are relatively cold. Under such circumstances, the engine must be turned slowly with the drips open, that the cylinder and piston may be warmed and all condensation blown out as fast as formed.

The principal causes of water in the steam are the boiler itself, the connecting piping, and the cold metal in the cylinder and valve chest.

112. Boiler Priming.—Boilers which supply wet steam are said to "prime" or "foam." The steam is rarely dry on leaving any boiler, but the amount of water should not exceed one or two per cent. The causes of priming in boilers are as follows:

(a) Insufficient steam room.

(b) Insufficient water surface for the disengagement of the steam.

(c) Improper location of the steam pipe.

(d) Dirty water.

(e) Overcrowding.

The first three causes are due to faults in design and tend to promote too violent ebullition and the splashing of water into the steam pipe. Upright boilers are particularly liable to be faulty in these respects. The use of muddy water, or that containing cylinder oil from the exhaust steam, sometimes causes priming. The remedy is obvious. One of the most common causes of wet steam is the forcing of a well-designed boiler beyond its capacity, with a mistaken sense of economy.

The subject of steam piping will be considered in detail in

Chapter XII., and it is only necessary to say here that all piping which conveys live steam should be thoroughly protected with a good grade of magnesia covering, and that it should slope in such a way that the water may flow in the same direction as the steam. Drop legs and separators should intervene between the boiler and the engine to protect the latter against any sudden flow of water, and there should be drips or bleeders at all low points in the system.

The steam should be brought to the cylinder in as dry a condition as possible, since further condensation is more rapid when the steam is initially wet.

113. Condensation in the Cylinder.—As long ago as 1769, James Watt, in the specifications for his patent on the steam engine, said in part:

"My method of lessening the consumption of steam, and consequently fuel, in fire engines, consists of the following principles:

"*First,* that vessel, in which the powers of steam are to be employed to work the engine, which is called the 'cylinder' in common fire engines and which I call the steam vessel, must, during the whole time that the engine is at work be kept as hot as the steam that enters it; first, by enclosing it in a case of wood or any other materials that transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and thirdly, by suffering neither water nor any other substance colder than the steam to enter or touch it during that time."

From that day to this, engineers have tried to keep the "steam vessel" as hot as the steam that enters it and with only partial success. The condensation of steam in the cylinder during admission and expansion has already been explained in Arts. 41-43, as well as the methods of determining the percentage of moisture from the indicator diagram.

The amount of condensation is influenced by certain conditions, such as the character and area of the clearance surface, the proportions of the cylinder, the speed of the engine, the average temperature of the cylinder walls, the rate of expansion and the quality of the entering steam. The less the area of cool surface exposed to the entering steam and the smoother that surface, the

less will be the condensation. This points to the advantage of a small clearance and of flat finished ends for both cylinder and piston. There should be no projections upon the head of the piston, and both it and the cylinder head should be turned flat and true. The clearance should be further reduced by bringing the valves as close to the cylinder as possible. (See the Corliss valve, Fig. 110.)

As regards the proportions of the cylinder, it may be said that the least surface for a given volume is secured by making the length equal the diameter. Since most of the condensation occurs during the early part of the stroke, long cylinders with small diameters would have an advantage in this respect.

Large engines have less difficulty with condensation than small ones, since the ratio of surface to capacity of cylinder is less. Engines having a high rotative speed suffer less from cylinder condensation, since there is less time for transfer of heat during each stroke. Mr. Willans' experiments on both condensing and non-condensing engines have shown a reduction from 20 per cent to less than 10 per cent of moisture at cut-off, caused by increasing the speed of the engine from about 120 to about 400 revolutions per minute. There was a corresponding decrease of from 12 to 15 per cent in the steam consumption.

An early cut-off and consequent high rate of expansion increase the amount of condensation on account of the extreme changes of temperature. Simple engines are limited to four or five expansions as an economical maximum. For cut-offs between one third and one fifth stroke, the economy is about the same, but if the cut-off is earlier than about one fifth, the condensation may offset any gain from increased expansion.

114. Steam Measurement from the Diagrams.—The method of showing the amount of dry steam present in the cylinder at any time during expansion, by means of a diagram, has been explained briefly in Arts. 41 and 43.

In order to develop this method more fully, as it would be used in testing an engine, a practical example will be solved. Let Fig. 132 be the indicator diagram of a 12 by 30 engine making 100 r.p.m. and using 2240 lb. of steam per hour. Let the clearance volume be 6 per cent of the piston displacement and

let the length of the diagram LH be 3.3 in. Assume the points B and C on the expansion line near the cut-off and release points respectively, and the point D on the compression curve near the beginning of compression. Points on the expansion line between B and C will show the quantity of dry steam present. At D the steam is assumed to be dry, since all moisture has usually been reëvaporated during the exhaust. Let the measured distances be as follows: $MB = 1.1$ in.; $NC = 3.2$ in.; $PD = 0.5$ in.

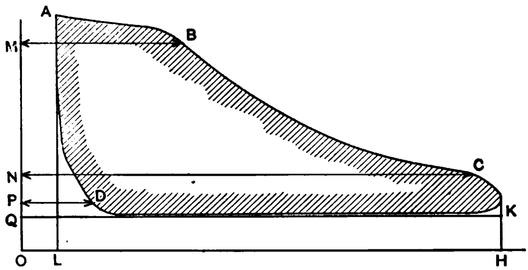


Fig. 132. Determining Dry Steam Present from Diagram.

Let the gage pressure at B be 81 lb.; at C, 23 lb.; at D, 8 lb. Let the barometer reading be 28.5 in.

Results: The piston displacement =

$$\frac{113 \times 2.5}{144} = 1.965 \text{ cu. ft.}$$

The volume per inch length of diagram =

$$\frac{1.965}{3.3} = 0.596$$

The volumes of steam present at the assumed points are therefore:

At B, $0.596 \times 1.1 = 0.656$ cu. ft.
" C, $0.596 \times 3.2 = 1.905$ " "
" D, $0.596 \times 0.5 = 0.298$ " "

The pressure corresponding to the barometer reading is

$$\frac{28.5}{2.036} = 14 \text{ lb.}$$

The following table gives the weight of steam present at the three points:

Point.	Absolute Pressure	Density.	Weight, lb.
B	95	.216	.1417
C	37	.0896	.1705
D	22	.055	.0164

The weight being greater at C than at B, shows the steam to be drier at release than at cut-off, as is usually the case. The weight at D is the amount of steam left in the clearance at the end of the stroke.

The weight of the mixture passing through the cylinder each stroke is:

$$\frac{2240}{60 \times 200} = 0.1867 \text{ lb.}$$

The amount in the cylinder to be accounted for during expansion is $0.1867 + 0.0164 = 0.2031$ lb.

The percentage of dry steam present at B is

$$\frac{0.1417}{0.2031} = 0.698$$

and at C,

$$\frac{0.1705}{0.2031} = 0.84$$

It is not necessarily true that all the loss shown by the preceding calculation is due to condensation. It is quite probable that some of it is occasioned by valve and piston leakage, especially the former.

115. Steam Jacketing.—The interchange of heat between the steam and the metal of the cylinder is confined to an extremely thin inside layer of the metal. It cannot be remedied by clothing the cylinder with non-conducting material, since the heat does not escape through the cylinder walls, but by the exhaust ports. It is, nevertheless, a loss because heat remains idle during the expansion and escapes during the release when no useful work is

done. There is also a certain amount of condensation due to the expansion of the steam doing work, which would occur even in a non-conducting cylinder, and this water has to be evaporated at the expense of the heat in the cylinder walls.

The condensation of steam during admission and expansion can be reduced by artificially raising the temperature of the cylinder walls. This increase of temperature will also cause an earlier re-evaporation of the film of water on the interior surfaces and thus increase the useful work. All the water which can be evaporated before release has a chance to do some work. The usual method of effecting this is by admitting live steam to cavities surrounding the entire body of the cylinder, including shell and heads.

It would also be desirable to admit steam to the inside of the piston and its rod, but this involves some mechanical difficulties. It is necessary that the steam jacket should be thoroughly protected from external radiation by non-conducting coverings.

The steam should be hot and dry on entering the jackets and all water of condensation must be promptly removed by drains and traps. In estimating the saving from the use of jackets, the steam thus condensed is added to that used inside the cylinder. The jacket steam is usually from 8 to 10 per cent of that used by the engine.

Steam jacketing is naturally most efficient where there is the greatest tendency to internal condensation, as in small engines, those running at a slow speed or those having a high degree of expansion.

Some of the experiments which have been quoted as showing a large economy from the use of jackets were made on very small engines, and are of little value commercially. Mr. Donkin's experiments, in which he shows a gain of 25 per cent from the use of the jacket, were made on a single cylinder 6 by 8 in. The engine used by Professor Reynolds was a small triple expansion, having cylinders 5, 8 and 12 in. in diameter, and he showed a saving of 17 per cent by the use of steam jackets.

There seems to be a great difference of opinion among steam users as to the value of this method of saving steam, the figures varying all the way from 2 per cent to 25 per cent of saving. The Institution of Mechanical Engineers of Great Britain has given

considerable attention to this subject. The reports made at various times by committees and individuals connected with that institution have shown an average saving from the use of jackets of from 12 to 15 per cent in condensing engines.

Experiments made in this country are not so favorable to their use. The tests made by Prof. J. E. Denton on the Pawtucket Pumping Engine* showed a saving of only 3 per cent. It is to be noticed, however, that in this engine the steam could not be cut out of the jackets on the heads of the cylinders. Further tests made by him three years later on a triple expansion pumping engine gave the following steam consumption under different conditions†:

Pressure in High Jacket.	Pressure in Int. and Low Jackets and Reheater.	Cut-off in Low Cylinder.	Dry Steam per hr. per i.h.p.
151 lb.	67 lb.	0.47	13.37
151 "	135 "	0.47	13.56
150 "	43 "	0.47	13.43
113 "	113 "	0.44	14.02
None	None	0.45	13.79
None	75 lb.	0.52	13.56
151 lb.	72 "	0.53	13.67
None	62 "	0.40	13.71

The jacket pressure in high cylinder was practically the same as boiler pressure. The pressure at cut-off was about 4 lb. less than this.

Professor Denton gives the following conclusions: "The steam economy is the same for the following conditions: (a) Any pressure from 43 to 131 in the intermediate and low jackets and receivers; (b) any pressure from 0 to 151 in the jacket of high cylinder."

He also found that slight variations in the cut-off in the three cylinders had no perceptible effect on the economy. He allows for 1 per cent of probable error in forming these conclusions. When it is considered that this engine ran at the low speed of 27.5 r.p.m., these results are the more remarkable.

* Transactions A. S. M. E., Vol. XI.
† Transactions A. S. M. E., Vol. XIV.

In tests made by Mr. Barrus and reported by him in his book on Steam Engine Tests, the cutting out of steam from the jackets and reheater in a compound engine had but little effect on the economy.

In summing up, it may be said that the steam jacket probably pays on the intermediate and low-pressure cylinders of large, slow-moving compounds, but that its efficiency on high-pressure cylinders or on high-speed simple engines is doubtful. Opinions differ widely and will continue to do so until there are more experimental data.

116. Superheating.—The nature of superheated steam and its graphic representation have already been discussed in Art. 42. Its use by the earlier experimenters was not altogether successful on account of its destructive action upon the cylinder lubricants and rod packings of that time, and it was soon abandoned in favor of high expansion.

The general introduction of high grade mineral oils for lubrication and of metallic packings for piston and valve rods has paved the way for the more general use of superheating. Like steam jacketing, it is most efficient where initial condensation is at its worst, as in slow-moving engines having high ratios of expansion. But unlike steam jacketing, it has an efficiency which is undisputed and which promises its more extended use in the future.

117. Specific Heat.—The lack of knowledge as to the specific heat of superheated steam has been a bar to theoretical investigation of its efficiency. For a long time, the specific heat was assumed to be constant and equal to 0.48, the value determined by Regnault.

Investigations by Griessmann, Lorenz and others have shown that, while this value is nearly correct for atmospheric pressure, the specific heat is not constant, but varies with the temperature at which superheating begins and with the amount of superheat.

The number of determinations made by the experimenters just named was not sufficient to indicate the law of variation. Only two facts could be regarded as definitely settled: (a) That C_p, or the specific heat at constant pressure, increased with the initial

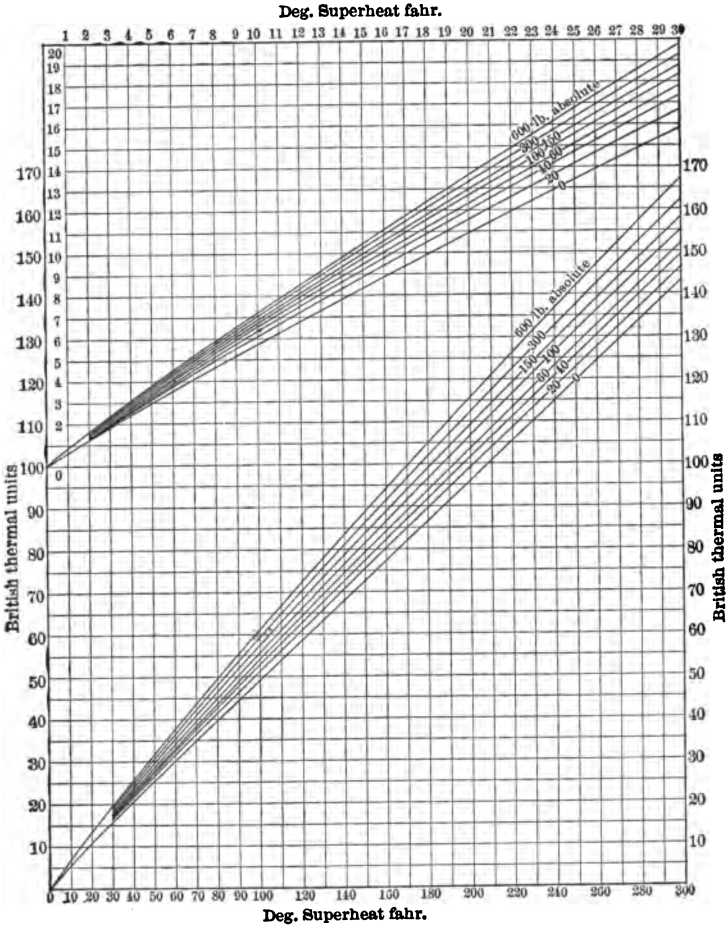


Fig. 133. B. t. u. Required to Superheat 1 lb. of Steam.

temperature, and (b) that it decreased with the amount of superheat.

More recent experiments conducted by Professor Thomas in this country and by Messrs. Knoblauch and Jakob in Germany were reported in papers read before The American Society of Mechanical Engineers in 1907.*

* Transactions A. S. M. E., Vol. XXIX.

Hundreds of tests were made by the former covering a range of pressures from 7 lb. to 500 lb. absolute and a range of temperature at each pressure from saturation to 270 deg. Fahr. superheat.

Fig. 133 shows one of the diagrams made by Professor Thomas to illustrate his experiments. From this diagram can be determined directly the number of B.t.u. required to heat steam at any pressure to any desired degree of superheat. Either the mean value of C or the value at any point can be taken off readily. For example, to heat steam at 60 lb. absolute from 0 to 150 deg. superheat will take 78.5 B.t.u. The mean value of C under these conditions is then, $78.5/150 = 0.523$.

Professor Thomas concludes that the specific heat of superheated steam varies with both pressure and temperature. It increases when the pressure of the steam increases and diminishes with an increase in the temperature. This variation occurs more rapidly when near the saturation point than is the case in conditions more remote from the saturation point.

118. Efficiency of Superheating.—The efficiency of superheated steam is slightly greater than that of the saturated steam from which it is derived, since the heat is received at a higher temperature. This difference is, however, so small, that the saving could not pay for the increased expense.

The chief advantage to be derived from the use of superheat is the prevention of initial condensation. The higher the degree of superheat, the drier will be the steam at cut-off, and it is possible to prevent cylinder condensation altogether in this manner and to deliver superheated steam during the exhaust.

In Professor Carpenter's tests on the White Motor Car, reported to the American Society of Mechanical Engineers,* he found the exhaust steam superheated 28 deg. after passing through two cylinders, the average superheat at entrance being 339 deg. Such extreme use of superheat is rare, and the most that is usually attempted is to secure dry steam at cut-off.

The degree of superheat necessary to do this depends upon local conditions, and no general rule can be formulated. Professor Ripper reports some experiments on a small Schmidt engine

* Transactions A. S. M. E., Vol. XXVIII.

in which he found that 7.5 deg. of superheat were necessary to remove each 1 per cent of moisture at cut-off. For example, if saturated steam showed 20 per cent of moisture at cut-off, 150 deg. of superheat were necessary to insure dry steam. The gain from such a rise of superheat can be best illustrated by an example.

Assume an engine to work on a Rankine cycle (Art. 38) between 100 lb. absolute and 15 lb. absolute, and assume the feed water temperature to be the same as that of the exhaust steam. Further assume the condensation at cut-off to be 20 per cent when dry saturated steam is used and that each 7.5 deg. of superheat removes 1 per cent of the moisture. It is required to find the thermal efficiency when using dry saturated steam and when using 75 deg. and 150 deg. of superheat.

Fig. 134 is the thermal diagram for this case.

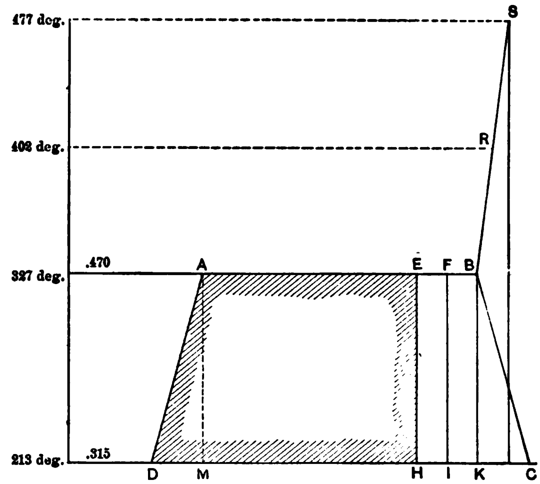


Fig. 134. Heat Diagram showing Efficiency with Superheated Steam.

(1) When saturated steam is used, the shaded area *AEHD* represents the work done and the total area under the line *DAEB*, the heat received.

$$\text{Heat received} = 1182 - 182.$$
$$= 1000 \text{ B.t.u.}$$

By calculation:

$$AB = \frac{884}{327 + 460} = 1.122 \text{ units.}$$

$$\begin{aligned} AF &= 0.9\ AB &&= 1.01 \\ AE &= 0.8\ AB &&= 0.898 \\ DM &= 0.470 - 0.315 &&= 0.155 \\ DK &= AB + DM &&= 1.277 \\ DI &= AF + DM &&= 1.165 \\ DH &= AE + DM &&= 1.053 \end{aligned}$$

$$\text{Area } AEHD = AM \times \frac{AE + DH}{2} = 114 \times 0.975 = 111 \text{ approx.}$$

$$\text{Efficiency} = \frac{111}{1000} = 0.111$$

(2) Seventy-five deg. of superheat would require 42.5 B.t.u. (See Fig. 133.) Heat received $= 1000 + 42.5 = 1042.5$. Assuming the superheated steam to expand adiabatically, this would be shown in the figure by a vertical line through R, and the efficiency would be but slightly more than that of the Rankine cycle $ABKD$.

The steam will really cool, as it enters the cylinder along the line RBF, condensing to the extent BF and leaving the proportion AF of dry steam at cut-off.

Area $AFID = AM \times \dfrac{AF + DI}{2} = 114 \times 1.087 = 124$ approximately.

Efficiency $= \dfrac{124}{1042} = 0.119$, a gain of about 7 per cent.

(3) 150 deg. of superheat would require 83 B.t.u. (from Fig. 133). Heat received $= 1000 + 83 = 1083$. The steam will now cool along the line SB and be dry at cut-off.

$$\text{Area } ABKD = AM \times \frac{AB + DK}{2} = 114 \times 1.2 = 137 \text{ approx.}$$

Efficiency $= \dfrac{137}{1083} = 0.127$, or a gain over the original efficiency of 14½ per cent.

It is of course understood that this is merely an illustrative example and that in actual practice the above figures might vary considerably.

119. Efficiency in Practice.—There is no lack of figures to prove the economy in steam and coal consumption due to the use of superheated steam. Professor Goodman states* that each 50 deg. of superheat will in ordinary cases reduce the steam consumption by about 10 per cent and that each 90 deg. of superheat will improve the economy 10 per cent.† It will be understood that the net saving of heat is less than the reduction in steam consumption, since the superheated steam contains more heat units per pound than the saturated steam.

Professor Ewing's tests on a cross-compound engine with a reheater between the cylinders‡ are quite conclusive. The steam in this case went first to the low-pressure reheater with 385 deg. superheat and reached the high pressure with a superheat of about 200 deg. The steam in the reheater was superheated 161 deg. The results of the trials with and without superheat were as follows:

	Saturated Steam.	Superheated Steam.
Indicated h.p.	124.8	184.25
Steam per i.h.p., per hour	17 2	10.4
Coal per i.h.p., per hour	2.1	1.3

These tests show the remarkable saving of 37 per cent of the coal used.

Some tests were made by Professor Jacobus in 1903 on a Rice and Sargent horizontal, cross-compound engine with a reheater between the cylinders.§

Two of the trials give the following results:

	Saturated Steam.	Superheated Steam.
Indicated h.p.	406.7	420.4
Steam per i.h.p., per hour	13.84	9.56
B.t.u. per i.h.p., per minute	248.2	203.7
Coal per i.h.p., per hour	1.497	1.257

* Cassier's Magazine, November, 1903.
† Cassier's Magazine, January, 1904.
‡ Cassier's Magazine, January, 1904.
§ Transactions, A. S. M. E., Vol. XXV.

The steam was used at a pressure of about 145 lb. at the throttle and the superheat was 374.5 deg. at the throttle and 141.4 deg. on entering the low pressure cylinder.

The most rational way of expressing the gain by superheating is on the heat unit basis.

In the experiments just quoted, the saving in heat units is 18 per cent, and is an example of what may be expected from a high degree of superheat. The saving in fuel is 16 per cent.

The use of superheated steam effects a saving in several ways. By far the most important of these is the prevention of condensation in the cylinder—explained in the preceding article. The same weight of superheated steam at the same pressure has a greater volume than saturated steam and can do more work. Undoubtedly, the absence of water in the cylinder reduces the leakage past valves and pistons. The smaller consumption of steam also means less boiler capacity and less grate area for the same horse power.

Superheated steam shows to the best advantage on single cylinder engines, which, by reason of their smaller size or slow speed, are subject to a large amount of cylinder condensation. In such engines, the use of superheat makes possible a much greater range of expansions, or in other words, makes it practicable to run the engine economically with either a light or a heavy load.

In compound engines, superheated steam allows of a larger cylinder ratio and larger rate of expansion, especially if a reheater is employed. The gain in economy is not so marked as in the case of simple engines. In fact, superheating may to a certain extent take the place of compounding, and it is probable that in the future two cylinder compounds with superheat will be preferred to triple expansion engines.

The experience of engineers points to the desirability of using a low degree of superheat, from 100 deg. to 150 deg. Fahr., unless engine and piping have been especially designed for the high temperature. A temperature of over 500 deg. Fahr. is apt to cause the rapid deterioration of brass or bronze, and by the destruction of the lubricant to produce more or less cutting and wear of valves and pistons. Even at a temperature of 500 deg., it will

be desirable to use metallic gaskets on the piping and special packings for the valve and piston rods.

The expansion of the piping is more troublesome with the higher temperatures and is apt to cause leaky joints. Any one who attempts to use superheated steam with ordinary piping, gaskets and packings, will usually regret the experiment; it is another case of putting new wine in old bottles.

The pipe covering must be of good quality and should protect flanges and other fittings as well as the pipe itself. With a pipe so covered, steam having a superheat of 100 or 150 deg. at the boiler may be expected to fall 1 deg. in temperature for every 6 or 7 ft. of pipe, in case a velocity of 6000 ft. per minute is maintained. The loss is usually greater than this, and is frequently over 50 deg. between the boiler and the engine. When the distance is very great, the effect of the superheat may be merely to deliver dry steam to the engine, instead of steam containing 8 or 10 per cent of water. It is doubtful if superheating the steam beyond the amount necessary to insure dry steam at cut-off is of any practical advantage.

With steam at a temperature of over 500 deg. Fahr., it will be desirable to use corrugated steel gaskets, non-conducting coverings of extra thickness on all piping, valve chests and cylinders, metallic rod packings, poppet valves and graphitic lubrication.

120. Superheating Devices.—Superheaters may be broadly classified as boiler heaters and independent superheaters. The former is attached to or connected with the boiler and is heated by the gases on their way to the chimney. For a small plant this type is more economical and is usually capable of giving a moderate degree of superheat. The independent superheater is entirely separate from the boiler plant, and the steam on its way from the boilers to the engine passes through it and is heated by an independent fire. This is the more suitable arrangement for a large plant, since one heater will serve for a considerable number of boilers. The economy is better since any desired degree of superheat can be attained independently of the temperature of the boilers.

When engines are running under a light load and an early cut-off, a higher degree of superheat is more desirable than when the

load is heavy and the cut-off late. The independent superheater permits of thus varying the temperature of the steam to suit the existing conditions.

The Schmidt superheater has been used extensively abroad and to a limited extent in this country. It can be used independently or in connection with a boiler, and consists of a series of flat coils of tube, one above the other, through which the steam circulates. The general course of the steam is from the top downward, in a direction opposite to that of the hot gases. To prevent burning of the lower coils, the cooler steam is passed through the two bottom ones, before passing to the top of the heater.

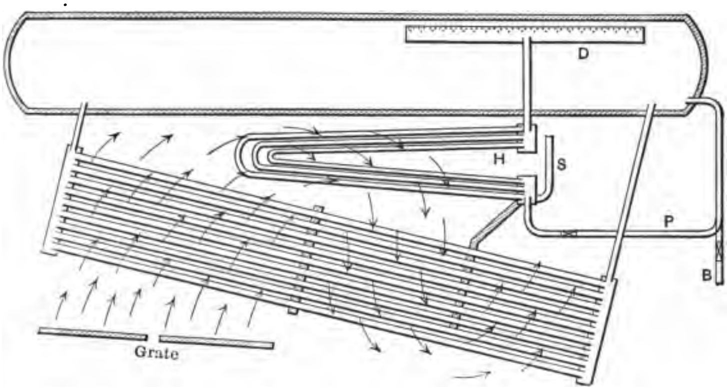


Fig. 135. Babcock and Wilcox Superheater.

Fig. 135 illustrates the Babcock and Wilcox superheater as applied to the boiler made by the same firm, a good example of the boiler type of heater. It is located in the path of the flue gases between the tubes and the drum and consists primarily of a series of U tubes connected at the ends by steel headers *H*. The steam is taken from a perforated dry pipe *D* located in the upper part of the drum and enters the upper header near the center; it then passes through the bent tubes into the lower header and is taken away by steam pipes *S* at either end; these latter unite in a tee above the drum, connecting with the steam main. A small pipe *P* provided with a blow-off *B* connects the

lower header with the water space in the drum so that the heater may be flooded while steam is being raised. This superheater is usually so proportioned as to give from 100 to 150 deg. of superheat.

Fig. 136, on the other hand, illustrates an independently fired superheater designed by the Stirling Boiler Company. The general design is similar to that of the Stirling water tube boiler, with drums at top and bottom connected by curved tubes.

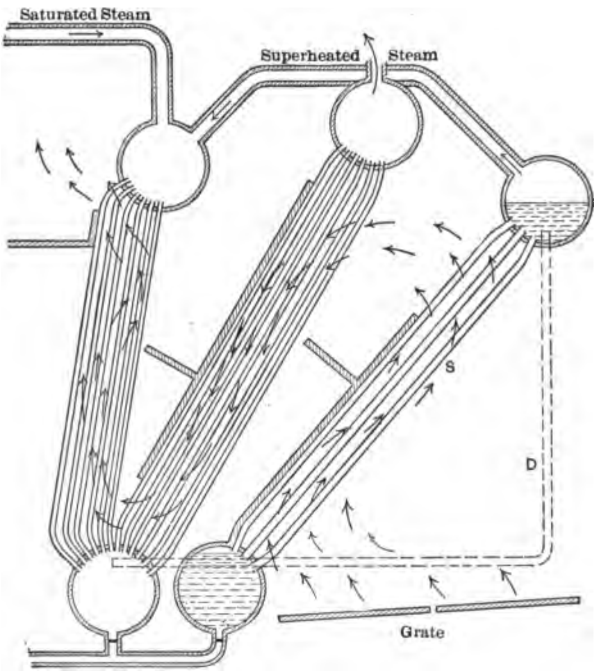


Fig. 136. Independently Fired Superheater.

The section *S* nearest the grate, including upper and lower drums and the connecting tubes, is used as an ordinary boiler, being filled with water to the usual level. The rear section, consisting of two upper drums, one lower drum and two sets of tubes, constitutes the superheater. The saturated steam from the boiler,

together with such steam as is generated by the front section, enters the rear drum at the top, passes down through the lower drum and up into the upper middle drum, from whence it is carried to the engines.

The circulation of the hot gases is opposite to that of the steam in the tubes. To promote circulation of the water in the boiler section, "down carrier" tubes *D* are carried from the upper to the lower drum in the brickwork at the ends. This superheater will furnish regularly a certain per cent of the steam used, and in emergencies can be used wholly as a boiler by flooding the superheater sections. Its size, compared with that of the boiler plant, will determine the degree of superheat, which may be anything desired up to 250 deg.

PROBLEMS.

1. A 14 x 18 simple, non-condensing engine makes 225 r.p.m. and has an initial pressure of 125 lb. by gage. The steam consumption by weighing tanks is 3600 lb. per hour. The clearance volume is 5 per cent of the piston displacement. The indicator card has a length of 3.6 in. and the following lengths and pressures by measurement. (See Fig. 132.) (The pressures are measured from the atmospheric line, the lengths from the end of diagram.)

Time.	Length.	Pressure.
Just after cut-off	1 in.	100 lb.
" before release	3.4 "	18 "
" after compression	0.3 "	12 "

(a) Make a table showing the volume, density and weight of all the dry steam present in the cylinder at each of these times.

(b) Find the weight of water and steam to be accounted for at each stroke.

(c) Find the dryness fraction of the steam at the beginning and end of expansion.

(d) Sketch diagram and show dry steam curve assuming no condensation.

2. Tests of a simple condensing engine, working between 120 lb. and 3 lb. absolute, show 34 per cent moisture in the steam at cut-off.

(a) Find the probable amount of superheat necessary to insure dry steam at cut-off.

(b) Find the thermal efficiency with and without superheat, assuming adiabatic expansion.

CHAPTER XI.

CONDENSERS AND HEATERS.

121. Condensers.—A condenser is a piece of apparatus used to condense the exhaust steam as it comes from the engine, by the application of cooling water. By using an air pump to remove the condensed steam from the apparatus, an exhaust pressure lower than that of the atmosphere may be continuously maintained. This process lowers the back pressure in the cylinders of the engine and increases the effective working pressure.

Condensers are broadly classified as surface and jet condensers. A surface condenser is one in which the exhaust steam and the cooling water are separated by some sort of a metal wall and do not mingle at any time. Surface condensers are more expensive, but are usually capable of maintaining a better vacuum than the jet condenser. They are generally used when the character of the cooling water is such as to render it unfit for use in the boilers, as is the case with sea water and some well water. The condensed steam is discharged by the air pump into a hot well, is purified and returned to the boilers, while the circulating water, as it is frequently called, is allowed to go to waste.

A jet condenser is one in which the steam and the cooling water come into direct contact and are discharged together by the air pump. This arrangement necessitates the use of a much larger air pump, but on the whole is usually cheaper than the surface condensing apparatus.

It has the further advantage that when the circulating water is pure, it dilutes the mixture coming from the cylinder and makes it easier to get rid of the oil. By running the whole into a large hot well, which overflows at the top, and taking the boiler feed from the bottom, comparatively little oil will be carried over.

122. Economy of Condensation.—By lowering the back pressure in the cylinder, the condenser increases the useful work done. The mean effective pressure in a simple non-condensing engine is usually from 40 to 50 lb. per sq. in. at normal load. The

use of the condenser adds 13 or 14 lb. to this effective pressure, or, we will say, 30 per cent.

This is not a net gain in power, for several reasons. It is probable that the condensation in the cylinder is increased by the lowering of the exhaust temperature, and therefore more steam will be used. The air pump and circulating pump consume a certain amount of steam which must be charged to the account. The cost of circulating water and the interest on the cost of the condensing apparatus are also items on the debit side. For example, suppose that an engine develops 225 i.h.p. when running non-condensing and uses 6750 lb. of steam per hour at a cost of $3.75 per hour. Assuming the mean effective pressure to be 42 lb. per sq. in. and that a condenser will lower the back pressure 14 lb., the increase in power will be one third or 75 i.h.p., and the engine will develop 300 i.h.p. with the same boiler pressure and grade of expansion as before.

Assuming the initial pressure to be 100 lb. gage, the difference between the initial and exhaust temperatures will have been increased about 120 deg., by using a condenser. This means more initial condensation in the cylinder and a larger consumption of steam.

If the moisture at cut-off was 20 per cent, it may be now 30 or 35 per cent, indicating an increase of 10 or 15 per cent in the steam consumption. Assuming an increase of 10 per cent from this cause and that the pumps use an additional 2 per cent, gives the total steam consumption at 7560 lb. per hour. From 15 to 20 lb. of circulating water would be needed per pound of steam, perhaps, 130,000 lb. or 2080 cu. ft. per hr. At 40 cents per thousand, this would amount to 83 cents per hour. The following summary shows the net gain:

NON-CONDENSING.

Cost of power per i.h.p. per hr. $= \frac{3.75}{225} =$ $0.0167

CONDENSING.

Total power developed,	300 i.h.p.
Total steam used per hour,	7560 lb.

Cost of steam per hour,	\$4.20
Cost of water per hour,	.83
Total cost per hour,	5.03
Cost of power per i.h.p. per hr. $= \frac{5.03}{300} =$	0.0168

In this example, the only gain is in the increased capacity of the engine and this could perhaps be secured more cheaply by a higher steam pressure. It does not pay as a rule to buy water at city prices for condensing purposes.

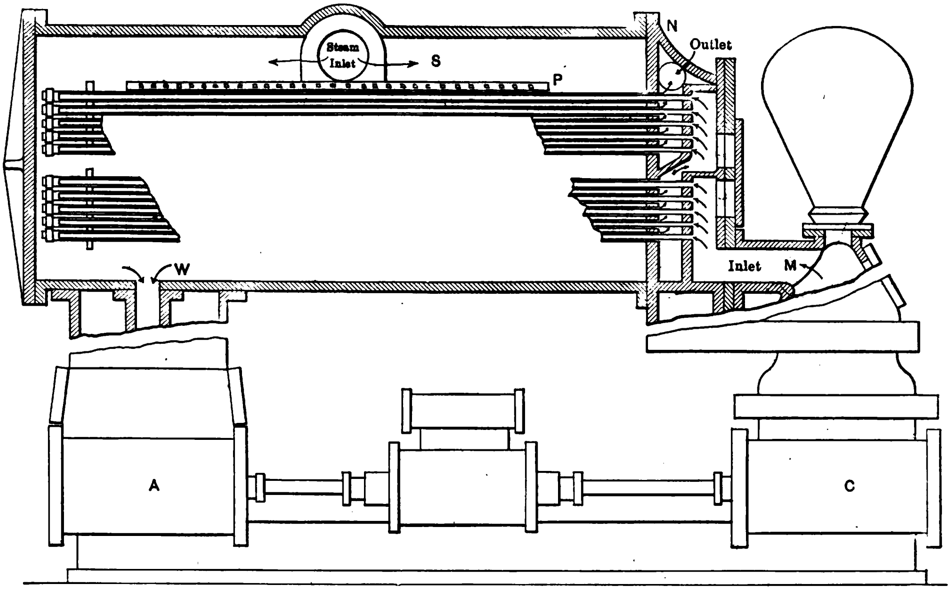


Fig. 137. Wheeler Surface Condenser.

Assuming in the preceding example that the circulating water is pumped from a river at the nominal cost of 2.5 cents per thousand, the total cost per hour is reduced to \$4.25 or only \$0.0142 per i.h.p. per hr., a saving of 15 per cent.

In general, condensing apparatus is economical for large rather than small engines, for superheated steam rather than for saturated, for compound engines rather than for simple.

123. Surface Condensers.—Fig. 137 shows the details of a modern surface condenser, with air pump *A* and circulating pump *C* attached. It consists of a cast-iron tank of a rectangular cross section, containing a large number of brass tubes through which the cooling water circulates. The exhaust steam enters at *S*, is diffused by the plate *P*, passes around the tubes and is withdrawn by the air pump at *W*.

The cooling water is forced in by the circulating pump at *M*, and following the course indicated by the arrows, emerges at *N*. In a double tube condenser like the one shown, the water passes back through the inner tube and returns in the annular space between the tubes. There are two advantages in this arrangement; the tubes are fastened only at one end and are free to expand and contract; the water returning between the inner and outer tubes is in the form of a thin film, which will receive heat rapidly and uniformly from the steam. Small condensers of this type have tube heads at each end and plain single tubes, the water passing through the lower bank and returning through the upper bank to the outlet.

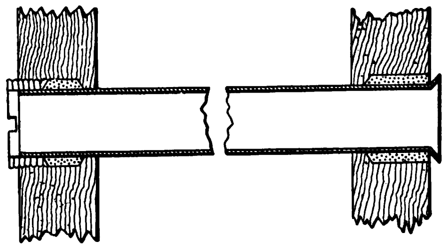


Fig. 138. Condenser Tube Joints.

Owing to the expansion and contraction of the tubes, there is difficulty in making a tight joint where they pass through the tube heads.

Fig. 138 illustrates two methods in common use, where wood pulp or some similar material is used for packing. The right end shows merely a flaring of the tube to prevent its working loose, while on the left end an improvement is made by using a screwed gland to compress the packing.

124. Size of Condenser.—The amount of cooling water

used by a condenser may be calculated as in the following example. Assume as in Art. 122 an engine of 300 h.p. using 7560 lb. steam per hr. and having an exhaust pressure of 2.5 lb. or a temperature of 135 deg. Fahr.

Assume the initial temperature of the circulating water to be 60 deg. and the final temperature of both to be 120 deg. The heat received by each pound of water is then 60 B.t.u., while that delivered by each pound of steam is 1035 B.t.u. Whence, $\frac{1035}{60} = 17.25$ lb. water per lb. steam. The condenser will then require 130,000 lb. circulating water per hour as given in Art. 122.

The area of condensing surface required will depend upon the material used for the tubes.

Let $S =$ area of condensing surface in sq. ft.
$W =$ weight of steam per hour.
$L =$ latent heat of steam.
$t =$ temperature of steam.
$t' =$ mean temperature of circulating water.
$k =$ thermal units transmitted per hour per square foot of condensing surface per degree difference of temperature.
$c =$ coefficient of efficiency.

Professor Whitham gives the value of k as 556.8 for brass (60 Cu., 4 Zn.) and finds $c = 0.323$ as the result of experiment. This gives the effective transmitting capacity of brass tubes as

$$ck = 180 \text{ B.t.u.}$$

The thermal units given up by the steam are approximately $= WL$ per hour. The heat capable of being transmitted by the brass tubes $= S \times 180(t - t')$. Equating these two expressions and solving for S

$$S = \frac{WL}{180\,(t-t')} \tag{34}$$

Applying this formula to the example just stated

$$S = \frac{7560 \times 1020}{180\,(135-90)} = 950 \text{ sq. ft.}$$

Undoubtedly, the value of c, and therefore of ck, depends somewhat upon the velocity of the circulating water.

The tubes are usually from ½ in. to 1 in. in diameter and about 1/16 in. thick.

In the foregoing example, if we assume the tubes as ¾ in. outside diameter, ⅝ in. inside diameter and 72 in. long, the inside surface of each tube will contain:

$$6 \times \frac{1.96}{12} = 0.98 \text{ sq. ft.}$$

or approximately one square foot. 950 tubes will accordingly be needed.

If the outer surface is used in the computation, the surface of each tube will be 1.18 sq. ft., and about 800 tubes will be sufficient.

125. Jet Condensers.—A good example of a modern jet condenser with independent air pump is shown in Fig. 139. The exhaust steam enters the pear-shaped condensing chamber at *A*, and meets there the cooling water which escapes from the slotted vertical pipe at *C*.

The mixture is drawn out by the air pump and discharged into the hot well through the pipe *J*. The quantity of cooling water is regulated by a valve on the supply pipe, while the hand wheel *E* is used to adjust the spray cone *D* in the most favorable position to insure a thorough mixing of the water and steam.

No circulating pump is used, as the air pump, by producing a partial vacuum in the condensing chamber, causes the water to flow in. The pump should not, however, be required to lift the water any considerable distance.

A much larger air pump is needed than with a surface condenser, since it is required to handle both the condensed steam and the circulating water. The simplicity and cheapness of the jet condenser commends it to steam users where clean water is available.

The use of a column of water instead of the air pump is illustrated in the siphon condenser, Fig. 140. The exhaust steam is carried to a height of 30 or 40 ft. above the engine and there meets a jet of cooling water, which has been raised to that height

by a circulating pump. A vacuum is formed in the combining chamber at *C*, and the condensed steam and water flow downward through the discharge pipe to the hot well below. The pressure of the atmosphere on the surface of the water in the hot well maintains a column of water in the discharge pipe, the

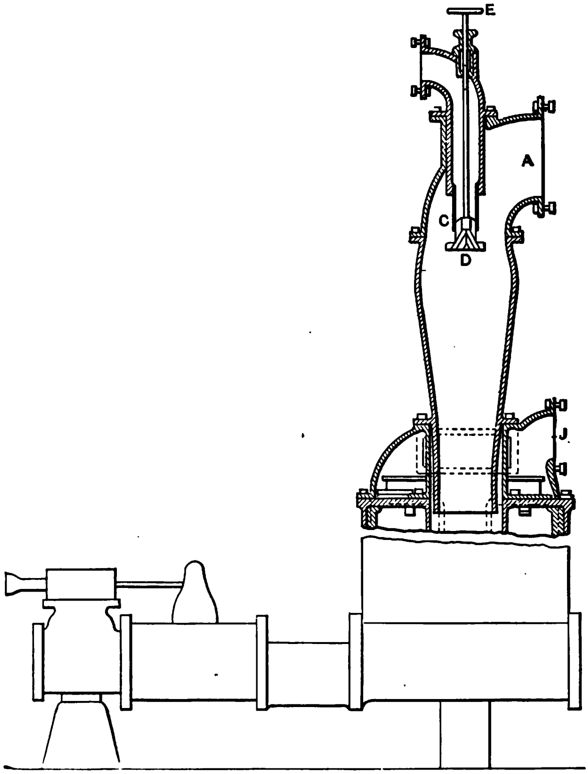


Fig. 139. Jet Condenser.

height of the column depending upon the degree of vacuum in the combining chamber. The injection water enters this chamber in the form of a hollow cone, surrounding the entering steam and so entraining the air as to cause a good vacuum.

In large condensers of this character, a "dry-air pump," so

called, is generally used to remove any air which may accumulate in the top of the condenser and so improve the vacuum. This pump may be small and comparatively quick acting, since it handles no water.

126. Cooling Towers.—The example which was worked in Art. 122 shows that the economy of condensing plants is small where the cooling water is obtained from city mains. Where an engine is so located that a supply of river or well water is not available, resource is had to artificial means of cooling the circulating water, so that it may be used over and over.

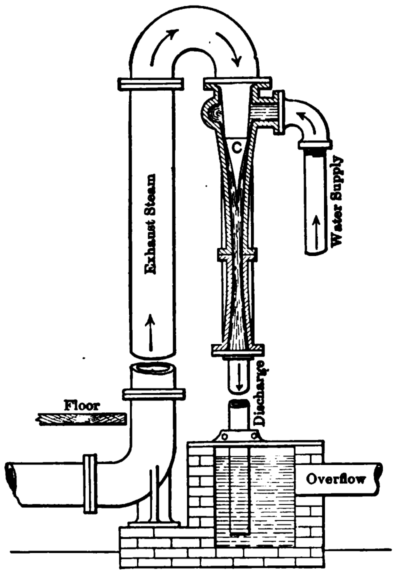


Fig. 140. Siphon Condenser.

Some establishments are provided with an artificial pond into which the air pump discharges and from which the supply of circulating water is taken; natural radiation is relied upon to cool the water.

Flat roofs have been utilized for the same purpose, the water flowing over the roofs in a thin film and parting with its heat

quite rapidly. In both of these systems the loss by radiation must be relatively large.

Another method is to pump the warm water over a series of shallow troughs arranged in the fashion of a wide stairway, so that the overflow from each trough descends into the one below, and the water is exposed in thin films to the natural winds or to air currents produced by blowers.

The Worthington cooling tower is one of the most modern devices for cooling condensing water and is in quite general use. It consists of a cylindrical steel tower over the suction tank, filled with ordinary drain tiles standing on end in tiers one above another. The tiles are so arranged as to break joints at the ends, and the water entering at the top flows in thin films over the exposed surfaces, where it is met by an ascending current of cool air from a blower. It is claimed that the loss by evaporation with this form of apparatus is no greater than the amount which is required for boiler feed. The amount of room taken up is small, and the only extra expense is that due to running the blower and to lifting the water to the top of the tower.

The Barnard cooling tower consists of a steel shell of rectangular section reinforced with angles and channels. Instead of tile, galvanized wire mats or nettings are used to break up the flow of water. These are suspended vertically inside the tower, and the water, distributed at the top by a series of perforated pipes, trickles down and over the wire netting. Fans are used for supplying an air current, as in the preceding type.

Mr. J. H. Vail in 1898 presented a paper before the American Society of Mechanical Engineers* which shows the economy of a condensing plant where cooling towers were employed. A twin tower of the Barnard type was installed, each section having a rated capacity such as to cool the circulating water needed to condense 12,500 lb. of exhaust steam per hour, from 132 deg. to 80 deg. Fahr. when the atmospheric temperature does not exceed 75 deg. Fahr., nor the humidity 85 per cent. The fans were operated directly by independent engines, so that the quantity of air used could be adapted to varying conditions.

To illustrate the performance of the apparatus at different

* Transactions, A. S. M. E., Vol. XX.

times of the year, the writer gives the following extracts from the log book:

	1898					
	Jan. 31	Feb.	June 20	July	Aug. 26	Nov. 4
TIME.	9 p.m.	8 p.m.	8 p.m.	8 p.m.	8 p.m.	5.35 p.m.
Temperature atmosphere, degrees Fahr.	30	36	78	96	85	59
Temperature condenser discharge to tower, degrees Fahr.	110	110	120	130	118	129
Temperature injection water from tower, degrees Fahr.	65	84	84	93	88	92
Number of degrees temperature was lowered by tower	45	26	36	37	30	37
Speed of fans, r.p.m.	36	0	145	162	150	148
Vacuum at condenser, inches	25½	26	25	24½	25½	25
Temperature boiler feed, degrees Fahr.	212	212	210	211	213	213

In a test made on a 20 and 36 by 42 in. compound engine in this plant, a total indicated horse power of 643.3 was developed, and of this amount 185.1 h.p. was done below atmospheric pressure. Deducting 27.25 h.p. required to run the air pump and the fans used for cooling, gives 157.85 h.p. as the net gain in power from the use of the condensing system, or 24½ per cent. The circulating water in this plant was pumped to the top of the tower by one cylinder of a twin air pump, the other being connected to the top of the condenser and used as a dry vacuum pump.

127. Heating Feed Water.—Although the subject of heating feed water belongs more properly in a treatise on steam boilers, it is desirable to consider it briefly here on account of its influence on the economy of the steam plant. Feed water heaters somewhat resemble surface condensers, as they usually consist of cast-iron receptacles filled with brass tubes, and since they serve to transmit heat from steam to water.

In the surface condenser, the main object is to cool the steam, while in the heater the raising of the temperature of the water is the principal function.

When the feed water is heated by the exhaust steam from the engine, the heater is a direct source of economy; when live steam

is used there is no direct gain, and the heater is used because the introduction of cold water into the boiler is objectionable. The gain in efficiency by heating with exhaust steam can be best illustrated by an example.

Assume that the boiler pressure is 100 lb. gage, and the exhaust pressure 2 lb. Neglecting all losses due to condensation and incomplete expansion, the thermal diagram will be as in Fig. 141.

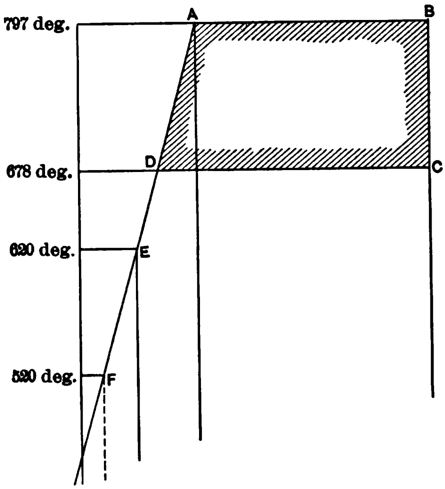


Fig. 141. Heat Diagram Showing Efficiency of Feed Water Heating.

Assuming first that the temperature of the feed water is 60 deg. Fahr., the total heat received from the boiler will be represented by the area under the line $FDAB$ and will be equal to

$$1185 - (60 - 32) = 1157 \text{ B.t.u.}$$

In case an exhaust steam heater is used for the feed water, assume the temperature to be raised to 160 deg. Fahr. The heat received from the boiler will now be the area under the line $EDAB$ or

$$1185 - (160 - 32) = 1057,$$

a saving of $\frac{100}{1157} = 8.65$ per cent.

128. Types of Heaters.—Feed water heaters may be divided into two general classes: open heaters and closed heaters, corresponding in a way to jet condensers and surface condensers.

An open heater is one in which the feed water and the steam come into direct contact. It usually consists of a horizontal cylindrical shell, containing a series of shallow pans one above the other. The water flows into the top pan and descends in thin films from one pan to another, coming into intimate contact with the exhaust steam which fills the chamber. If the water is raised to sufficiently high temperature, much of the scale forming material is deposited on the pans. These can be removed from time to time and cleaned.

This form of heater is well adapted for use with feed water which contains a large per cent of impurities. Such a heater cannot be used under pressure, and is therefore located on the suction side of the feed pump. This necessitates the use of a pump suitable for handling hot water.

The closed heater is located between the feed pump and the boiler, and is therefore under full boiler pressure. It is similar to a surface condenser in principle, having brass or steel tubes for separating the water from the steam.

The feed water may flow through the tubes and the steam outside or vice versa. Fig. 142 shows a diagram of the Otis tubular heater in which the steam passes through the tubes. The latter are of seamless brass, are curved at the bottom to allow for expansion and connect there with a chamber where the condensed steam and oil collect and may be blown off. The exhaust steam enters at *A,* passes down through the tubes to the separator, then up to the outlet at *B*.

The cold water enters at *C* and fills the space around the tubes, passing out at *D*. The lower part of the heater forms a settling chamber for the scale and mud in the water and is provided with a blow-off pipe and with hand holes for cleaning.

The principal advantage of having the water outside of the tubes is that much of the scale is precipitated on account of the low velocity of the water and that this scale does not clog up the tubes.

When a heater is used in connection with a condensing engine,

the exhaust steam passes through the heater on its way to the condenser, and therefore the amount of circulating water required is less. The usual allowance for heating surface is $\frac{1}{3}$ sq. ft. for each nominal horse power.

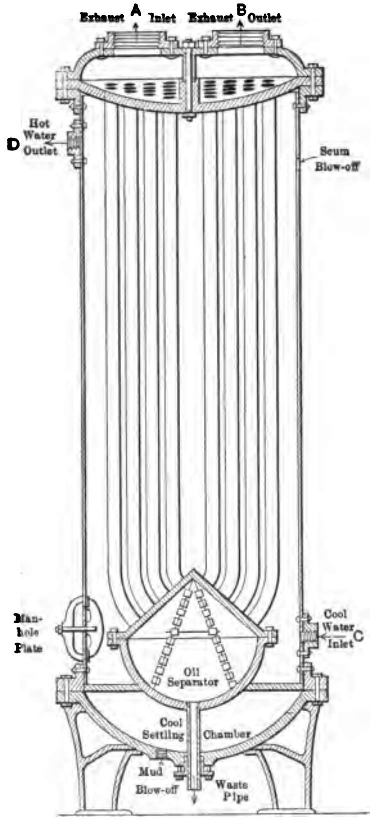


Fig. 142. Feed Water Heater.

The term economizer has come to be applied to feed water heaters which are located in the path of the furnace gases, the latter taking the place of steam as a heating agent. This amounts to practically the same thing as heating in the boiler itself.

129. Injectors.—Although the injector is primarily an instrument for pumping water into a boiler, it is also an efficient means of heating the feed water and demands notice in this connection. It is also one of the most interesting examples of the conversion of heat into mechanical energy, without the interposition of a piston or other moving piece.

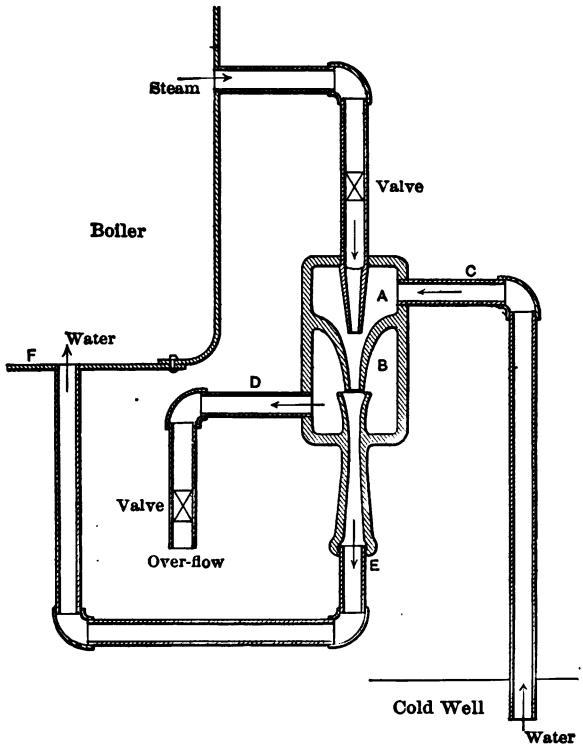


Fig. 143. Showing Principle of Injector.

The principle of the injector may be most easily explained by reference to an ideal diagram, Fig. 143. Steam from the boiler enters the injector through the steam nozzle at A. The partial vacuum caused by the flow of the steam causes the water in the well to rise in the pipe and enter the injector at C. B is the combining tube where the water mixes with the steam and is forced through the delivery tube at E, and so on into the boiler at F.

When the injector is first started, the jet of steam, entering B at a high velocity, carries with it the air from the surrounding chamber, and passes out of the overflow at D. The water, entering at B is also carried along with the steam and flows out at D. The valve in the overflow pipe is then closed and the water is forced out at E and into the boiler.

130. Theory of the Injector.—Stated briefly, the theory of the action of the injector is this: The steam, passing from the high pressure in the boiler to the low pressure in the injector chamber, expands suddenly so that a portion of its heat energy is converted into energy of motion. Condensed by contact with the cool water, it gives up its latent heat to warm the water, but retains enough of its kinetic energy to carry the water along with it against the boiler pressure.

Let p = gage pressure of the steam in the boiler.
q = heat of liquid at above pressure.
L = heat of evaporation.
x = quality of steam entering injector.
W = pounds of water lifted per pound of steam.
t_1 = initial temperature of water, degrees Fahr.
t_2 = final temperature.
q_1 and q_2 = corresponding heats of liquid.
S_1 = height of lift in feet from well to injector.
S_2 = height of lift from injector to water level in boiler.
v = velocity in feet per second of water leaving injector.

The heat lost by the steam expressed in foot pounds is

$$JH = J\,(x\,L + q - q_2)$$

The heat given to water is:

$$\begin{aligned} Jh &= JW\,(q_2 - q_1) \\ &= JW\,(t_2 - t_1) \text{ nearly.} \end{aligned}$$

The work of lifting the water to the combining tube is

$$WS_1 \text{ foot pounds}$$

and the work of lifting further to the boiler level is

$$(W+1)S_2 \text{ foot pounds.}$$

The kinetic energy of the water as it leaves the injector is

$$\frac{(W+1)\,v^2}{2g} \text{ foot pounds.}$$

This energy must be sufficient to lift the water the remaining distance and force it into the boiler.

A pressure of p pounds per square inch is equivalent to a head in feet of $\frac{144p}{62.4} = 2.31p$.

Therefore,

$$\frac{(W+1)\,v^2}{2g} = (W+1)\ (2.31p+S_2)$$

or

$$\frac{v^2}{2g} = 2.31p+S_2 \qquad (35)$$

i.e. velocity head = pressure head + lift.

If v is less than the value indicated by equation (35), the injector will fail to work. If it is greater, the water will have some residual velocity on entering the boiler. No allowance has been made for the friction in the pipe, which is considerable.

Weight of Discharge.—The heat energy given up by the steam must raise the water, heat it from t_1 to t_2 and impart to it the velocity v; hence, if no allowance is made for radiation losses:

$$J(xL+q-q_2) = JW(q_2-q_1)+WS_1+\frac{(W+1)\,v^2}{2g} \qquad (36)$$

If we neglect the weight of the steam in the above equation, we have for an approximate value:

$$W = \frac{J(xL+q-q_2)}{J(q_2-q_1)+S_1+\frac{v^2}{2g}} \qquad (37)$$

Velocity of Discharge.—To determine the value of v, we will assume the velocity of the steam before striking the water to be u feet per sec.

The value of u will vary from 1450 to 1520 according to the boiler pressure and may be assumed as 1500 for ordinary calculation. See Art. 131—133.

Then, as the momentum after impact is the same as before, we will have (neglecting the momentum of suction water):

$$u = (W+1)\,v$$

or

$$v = \frac{u}{W+1} = \frac{1500}{W+1} \tag{38}$$

The mechanical efficiency of the injector is small, since nearly all of the heat of the steam is expended in raising the temperature of the water.

The last two terms in the denominator of equation (37) are therefore relatively small and may be neglected in getting an approximate value of W. With this value of W, an approximate value of v is obtained in equation (38). Substituting this value of v in either (36) or (37) will give a value of W which is accurate enough for ordinary purposes.

Example.—Assume the case of an injector which is to pump water into a boiler against 120 lb. gage pressure, lifting it from a well 4 ft. below the injector and raising it 8 ft. further to the water level of the boiler. The temperature of the water in the well is 60 deg. and of the mixture 170 deg. Fahr.

The heat given up by the steam is

$$778(867.3 + 321.4 - 138.5) = 778 \times 1050 = 816{,}900 \text{ ft. lb.}$$

Assuming the entering steam to be dry, the heat given to each pound of water is

$$778(170 - 60) = 85{,}580 \text{ ft. lb.}$$

The approximate value of W will be

$$\frac{816{,}900}{85{,}580} = 9.55$$

Using this value of W, the velocity after combining will be

$$v = \frac{1500}{9.55+1} = 142 \text{ ft. per sec.}$$

The kinetic energy of each pound of the mixture is now

$$\frac{v^2}{2g} = \frac{142 \times 142}{64.4} = 314 \text{ ft. lb.}$$

Using formula (37) for W,

$$W = \frac{816,900}{85,580+4+314} = 9.51, \text{ a sufficiently close approximation.}$$

The head equivalent to a pressure of 120 lb. per sq. in. is

$$2.31 \times 120 = 277 \text{ ft.}$$

Adding to this the lift of 8 ft. gives 285 ft. as the necessary velocity head.

Subtracting this from the available head leaves $314 - 285 =$ 29 ft. as the head left to overcome friction. If we neglect friction, the water will enter the boiler with a velocity $=$

$$\sqrt{64.4 \times 29} = 43.2 \text{ ft. per sec.}$$

The work in foot pounds done by the heat energy of the steam may be thus classified.

From steam,	816,900 ft. lb. =	100 per cent
Heating water,	813,800 ft. lb. =	99.4 per cent
Mechanical work,	3100 ft. lb. =	0.6 per cent

The radiation loss is sometimes as high as 5 or 10 per cent, as determined by experiment. As a feed water heater and pump combined, the injector is economical, but for general pumping purposes it is a very wasteful device.

131. Automatic Injectors.—The injectors most used in stationary work are of the automatic or self-starting class. Fig. 144 shows an injector of this type. R is the steam nozzle and S the combining tube, while O is the regular discharge to the boiler. The overflow chamber has a check valve P opening outwards.

When steam is admitted to the injector, the steam and water escape through the check valve, but as soon as a vacuum is established, the valve is closed by the pressure in the overflow pipe, and the water is forced through the regular discharge into the boiler. If from any cause the discharge is stopped, the check valve again opens and remains open until normal conditions are again restored.

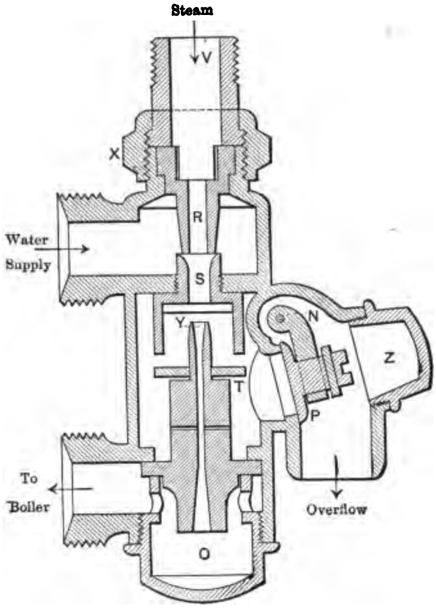


Fig. 144. Section of Injector.

Problems.

1. Determine the economy of using a jet condenser which shall maintain a vacuum of 25 in. of mercury with an expenditure of 20 lb. of cooling water per pound of steam.

The engine develops 400 i.h.p. when running non-condensing, and uses 10,800 lb. steam per hour at a cost of $4 per hour. The mean effective pressure at full load is 50 lb. per sq. in. and the usual back pressure is 2 lb. above the atmosphere.

2. Determine the number of square feet of condensing surface needed for a surface condenser for the engine in Problem 1, if the cooling water is raised from 50 to 100 deg. Fahr.

3. Determine the number of brass tubes 6 ft. long for the above condenser, and the velocity of the water in feet per minute, if it passes through three banks of tubes.

The tubes are to be 1 in. outside diameter and $\frac{1}{16}$ in. thick.

4. If feed water is heated from 50 deg. to 170 deg. Fahr., what is the percentage of saving, the boiler pressure being 110 lb. gage? If the boiler evaporates 3600 lb. water per hour, how many brass tubes 4 ft. long and 1.25 in. in diameter will be needed in the heater?

5. An injector takes water at a temperature of 75 deg. from a well and delivers it at a temperature of 140 deg. against a boiler pressure of 90 lb. gage. Assuming the velocity of the steam to be 1450 ft. per second and neglecting the lift, find the amount of water pumped per pound of steam and the residual velocity.

CHAPTER XII.

PIPING AND FLOW OF STEAM.

132. General Laws.—The flow of steam through a pipe or orifice is caused by a difference of pressure, just as flow of heat is due to difference of temperature, or electric current is due to difference of potential.

If losses due to radiation and conduction are neglected, the energy in the steam will remain the same, since any apparent loss by friction will reappear in rise of temperature or in re-evaporation.

If there be friction on the sides of the pipe or orifice, a part of the kinetic energy due to velocity will be changed back to heat, and the expansion will not be adiabatic. If the friction is excessive, practically all of the energy may be thus absorbed, and the final velocity will be small, as is sometimes the case when the orifice is minute.

Let p_1 and T_1 represent the pressure and temperature in the reservoir and q_1 and L_1 the corresponding heats.

Let $p_2T_2q_2L_2$ represent the conditions in the pipe.

Suppose that the steam in the reservoir is x_1 per cent dry and in the pipe x_2 per cent dry.

We will neglect the velocity of flow in the reservoir and call the velocity in the pipe u in feet per second. Then will the mechanical kinetic energy of a pound of steam in the pipe be $\dfrac{u^2}{2g}$.

The general expression for the heat energy in wet steam is

$$JH = J(q + xL)$$

If no energy is lost by conduction and radiation we shall then have:

$$J(q_1 + x_1L_1) = J(q_2 + x_2L_2) + \frac{u^2}{2g}$$

(neglecting the volume of the condensed steam)

and

$$\frac{u^2}{2g} = J(q_1 - q_2 + x_1L_1 - x_2L_2).$$

Fig. 145 shows the changes of heat and entropy in the process, the shaded area corresponding to $\frac{u^2}{2g}$.

To determine the relations between x_1 and x_2 we must remember that if no heat is lost or gained by conduction the change is adiabatic and therefore the entropy remains the same, or

$$x_1\phi_1+\theta_1 = x_2\phi_2+\theta_2$$
$$x_1\phi_1-x_2\phi_2 = \theta_2-\theta_1$$

But $$\phi = \frac{L}{T} \quad \text{and} \quad \theta_2-\theta_1 = \log_e \frac{T_2}{T_1}$$

$$\frac{x_1L_1}{T_1} - \frac{x_2L_2}{T_2} = \log_e \frac{T_2}{T_1}$$

$$x_2L_2 = \frac{T_2}{T_1}\, x_1L_1 - T_2\log_e \frac{T_2}{T_1}$$

Substituting this value in the equation of energy and reducing, there results

$$\frac{u^2}{2g} = \mathrm{J}\left\{ q_1-q_2+x_1L_1\left(1 - \frac{T_2}{T_1}\right)+T_2\log_e \frac{T_2}{T_1}\right\} \qquad (39)$$

From this we may determine u, the velocity of the steam.

In Fig. 145, let B denote the condition of the entering steam having a quality $\frac{AB}{AE}$, and let EF be the saturation line. If the steam expands adiabatically without friction until it reaches some lower temperature, as at C, the shaded area $HABC$ will represent the heat converted into kinetic energy and the quality of the steam will be $\frac{HC}{HF}$; but if some of the energy is converted back into heat by friction, the change will be indicated by some other line to the right of BC, such as BD and the steam will be dryer than before, its quality now being $\frac{HD}{HF}$. It contains more latent heat than before, as shown by the rectangle $CDNM$, and this amount must be subtracted from the shaded area to give the quantity representing the kinetic energy. If BD were the curve of constant heat, the area under CD would be equal to the shaded

area and the steam would have no velocity, all of the heat having been restored by friction. If D falls to the right of the saturation line EF, as it might were the steam initially dry, the steam would be superheated by friction as is the case in throttling calorimeters (see Art. 142).

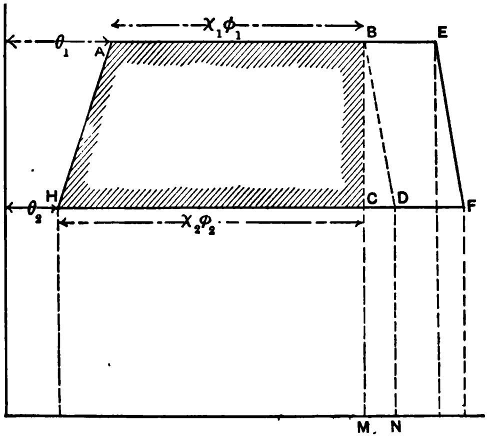


Fig. 145. Heat Diagram for Flow of Steam.

133. Flow Through Orifices or Short Tubes.—Napier's formula for the weight of steam flowing from an orifice is derived from the results of experiments and is closely correct when p is equal to or greater than five thirds of the atmospheric pressure.

Let p_1 = absolute initial pressure in pounds per square inch and A = the area of the orifice in square inches. Then the discharge in pounds per second will be

$$W = \frac{p_1 A}{70} \tag{40}$$

To find velocity from Napier's formula,

Let d = weight of one cubic foot of steam at exit.

Q = volume in cubic feet per second.

u = velocity in feet per second at exit.

Then $Q = \dfrac{A\,u}{144}$ and $W = Qd = \dfrac{A\,u\,d}{144}$

Therefore $\dfrac{A\,u\,d}{144} = \dfrac{p_1 A}{70}$ or $u = \dfrac{144\,p_1}{70\,d} = \dfrac{2p_1}{d}$ approx. (41)

If the steam flows through an orifice, the pressure at the orifice will not be less than about 0.57 of the initial pressure. Under these circumstances, the velocity at the orifice will be from 1400 to 1500 feet per second, depending upon the value of p, the initial pressure.

For example, assume an absolute pressure of 115 lb. per square inch in the boiler and assume the boiler steam to be dry. The pressure at the orifice will then be

$$p_2 = 115 \times 0.57 = 65.5 \text{ lb.}$$

$$\frac{u^2}{2g} = 778\ (H_1 - q_2 - x_2L_2)$$

$$x_2L_2 = \frac{T_2}{T_1}\ L_1 + T_2 \log_e \frac{T_1}{T_2}$$

$$T_1 = 338 + 460 = 798$$

$$H_1 = 1185$$

$$L_1 = 876.3$$

$$T_2 = 298 + 460 = 758$$

$$q_2 = 267.7$$

$$x_2L_2 = 0.95 \times 876.3 + 758 \log_e\ 1.053 = 832.5 + (758 \times 0.05)$$

$$= 870.4$$

$$\frac{u^2}{2g} = 778\ (1185 - 267.7 - 870.4) = 778 \times 47 = 36,570$$

$$u = \sqrt{2,355,000} = 1535 \text{ ft. per sec.}$$

The velocity in the preceding example, as determined by Napier's formula, would be

$$u = \frac{144 \times 115}{70 \times 0.153} = 1546 \text{ ft. per sec.}$$

If the steam expands further, as in the flaring nozzle of a steam turbine, the velocity will increase, but the weight of steam discharged will depend upon the velocity at the contracted portion. On the other hand, if the steam is warmed by friction in passing through the orifice or nozzle, the velocity will be less and the steam will be drier, as explained in Art. 132.

For example, the exit steam in the problem of Art. 133 was found to have the value

$$x_2L_2 = 870.4$$

But L_2 for dry steam at 65.5 lb. pressure is 905.2

Therefore,

$$x_2 = \frac{870.4}{905.2} = 0.962$$

Suppose that the steam was found to be 98 per cent dry at exit (corresponding to point D in Fig. 145), it would then contain $0.98\ L_2 = 0.98 \times 905.2 = 887$ B.t.u. instead of 870.4.

Substituting this value in our formula, we have,

$$\frac{u^2}{2g} = 778\ (1185-267.7-887)$$

Reducing, $u = 1232$ ft. per sec.

134. Flow of Steam in Long Pipes.—The formulas which have just been given have little to do with the flow of steam in pipes between the boiler and the engine. The velocity of the steam in such case will depend entirely on the amount used by the engine and the size of the pipe. The fall of pressure will be governed by the velocity of the steam, the size of pipe, the material of which it is made, and the number of elbows, tees and valves, very much as in the case of a water pipe.

The condensation of steam will depend on its temperature, the size of pipe and the nature of the covering, and will be practically independent of the fall of pressure.

The formula most commonly used for the weight of flow of steam in long pipe is as follows:

$$W = 87\sqrt{\frac{w\ (p_1-p_2)\ d^5}{l\left(1+\frac{3.6}{d}\right)}} \qquad (42)$$

Where

$W =$ weight of steam per minute.
$w =$ weight of steam per cubic foot.

$p_1 =$ initial pressure.
$p_2 =$ final pressure.
$d =$ diameter of pipe in inches.
$l =$ length of pipe in feet.

This formula is partly empirical and is similar to the one used for flow of water in pipes. The latter may be found in any standard work on hydraulics. The losses due to elbows and valves can be allowed for by substituting equivalent lengths of pipe.

The formula for such lengths is

$$l = kd-\left(1 + \frac{3.6}{d}\right) \tag{43}$$

The values given in "Steam" for the constant are

Elbows, $k = 76.$
Globe valves, $k = 114.$

Carpenter's experiments, however, give much larger values for k, nearly seven times those quoted. Gate valves offer little resistance to the flow of steam.

Estimates of this kind must be regarded as approximations, and the friction of pipe and of fittings may vary widely from that assumed in the formulas. Formula (42) may be used in estimating the probable drop of pressure for a given discharge by solving for $(p_1 - p_2)$, thus:

$$p_1 - p_2 = \frac{W^2 l\left(1+\frac{3.6}{d}\right)}{7569 w d^5} \tag{44}$$

or solving for the diameter,

$$d = 0.167 \sqrt[5]{\frac{W^2 l\left(1+\frac{3.6}{d}\right)}{(p_1 - p_2)w}} \tag{45}$$

These formulas are all based on a value of the coefficient of friction f, which is proportional to $\left(1 + \frac{3.6}{d}\right)$

Simpler equations are based on the assumption that f is constant and take the following form:

$$W = 54\sqrt{\frac{w(p_1-p_2)d^5}{l}} \tag{46}$$

$$p_1-p_2 = \frac{W^2 l}{2916\, wd^5} \tag{47}$$

$$d = 0.201\sqrt[5]{\frac{W^2 l}{(p_1-p_2)\, w}} \tag{48}$$

For a discussion of the various formulas on this subject, see the article by Prof. G. F. Gebhardt in "Power" for June, 1907.

135. Carpenter's Experiments.—Prof. R. C. Carpenter reported some experiments on the loss of pressure in pipes in 1899. Reference is here made to Vol. XX. of the Transactions of the American Society of Mechanical Engineers, where the report is given in full. The experiments were made on pipes varying in diameter from 1 to 3 in. and of various lengths.

As a result of these experiments and of a discussion of those made in 1892 by M. Ledoux, Professor Carpenter derives a formula identical with (42) Art. 134, except that the co-efficient of the radical is 87.45.

The following table from "Steam" was calculated from formula (42). The flow for other lengths and pressure drops can be easily calculated from this table.

TABLE I.

Table of Flow of Steam Through Pipes.

Initial Pressure by Gage Pounds per Square Inch.	Diameter of Pipe, inches. Length of each = 240 diameters.													
	¾	1	1½	2	2½	3	4	5	6	8	10	12	15	18
	Weight of Steam per Minute, in pounds, with one pound loss of pressure.													
1	1.16	2.07	5.7	10.27	15.45	25.38	46.85	77.3	115.9	211.4	341.1	502.4	804	1177
10	1.44	2.57	7.1	12.72	19.15	31.45	58.05	95.8	143.6	262.0	422.7	622.5	996	1458
20	1.70	3.02	8.3	14.94	22.49	36.94	68.20	112.6	168.7	307.8	496.5	731.3	1170	1713
30	1.91	3.40	9.4	16.84	25.35	41.63	76.84	126.9	190.1	346.8	559.5	824.1	1318	1930
40	2.10	3.74	10.3	18.51	27.87	45.77	84.49	139.5	209.0	381.3	615.3	906.0	1450	2122
50	2.27	4.04	11.2	20.01	30.13	49.48	91.34	150.8	226.0	412.2	665.0	979.5	1567	2294
60	2.43	4.32	11.9	21.38	32.19	52.87	97.60	161.1	241.5	440.5	710.6	1046.7	1675	2451
70	2.57	4.58	12.6	22.65	34.10	56.00	103.37	170.7	255.8	466.5	752.7	1108.5	1774	2596
80	2.71	4.82	13.3	23.82	35.87	58.91	108.74	179.5	269.0	490.7	791.7	1166.1	1866	2731
90	2.83	5.04	13.9	24.92	37.52	61.62	113.74	187.8	281.4	513.3	828.1	1219.8	1951	2856
100	2.95	5.25	14.5	25.96	39.07	64.18	118.47	195.6	293.1	534.6	862.6	1270.1	2032	2975
120	3.16	5.63	15.5	27.85	41.93	68.87	127.12	209.9	314.5	573.7	925.6	1363.3	2181	3193
150	3.45	6.14	17.0	30.37	45.72	75.09	138.61	228.8	343.0	625.5	1009.2	1486.5	2378	3481

136. Allowable Velocity.—The size of steam pipe for a given engine is frequently determined on the basis of a certain maximum allowable speed of steam. The more common rule is as follows:

$$\text{Area pipe} = \frac{\text{area piston} \times \text{piston speed in ft. per min.}}{6000}$$

This rule allows a steam velocity of 6000 ft. per minute and usually makes the area of steam pipe about one tenth that of the piston. Since in most engines the steam is only admitted for one sixth to one fourth of the stroke, the flow of steam is intermittent and causes a fluctuation of pressure in the pipe near the engine.

Marine practice frequently calls for a live steam velocity of from 8000 to 10,000 ft. per minute and an exhaust steam velocity of 6000 ft. English engine builders sometimes use live steam velocities of 12,000 ft. per minute. The use of higher pressures and of superheat favors smaller sizes of pipe, to reduce radiation.

The greater density of high pressure steam is also an argument for the use of smaller piping. Mr. E. H. Foster, in a paper read before the American Society of Mechanical Engineers, recommends a velocity of from 6000 to 8000 ft. per minute for superheated steam. He shows that although the loss of heat per square foot of surface per degree difference of temperature is greater at the higher velocities, the percentage of heat lost is less on account of the greater quantity of steam passed.

The loss of temperature per 100 ft. of pipe is reduced more than 50 per cent by increasing the velocity from 2000 to 6000 ft. per minute. For example, consider saturated steam at a pressure of 90 lb. gage, and steam at 150 lb. gage with 100 deg. superheat. The temperature of the former is 331 deg. Fahr. and of the latter 466 deg. Assuming the temperature of the room to be 60 deg., the loss by radiation would be proportional to 271 and to 406 in the two cases, an excess of 33 per cent for the hotter steam.

On the other hand, the hotter steam, notwithstanding its superheat, would have a weight per cubic foot of 0.324 lb. as against 0.237 lb. for the steam of the lower pressure, so that a pipe of

three fourths the capacity would serve as well to carry the same weight of the hotter steam.

In order to check the fluctuations due to the use of smaller pipe, some builders provide a receiver near the engine, having a capacity about three times that of the high pressure cylinder and use the customary size of pipe next to the cylinder, while the remainder of the pipe is two sizes smaller.

For example, a medium speed four-valve engine, built by a well-known firm, has a cylinder 24 x 36 in. and a speed of 125 revolutions per minute. The diameter of the steam pipe is 9 in., having an area in cross-section of 62.7 sq. in.

Since the area of the piston is 452.4 sq. in., and the average piston speed is 750 ft. per minute, the average speed of the steam entering the cylinder would be about

$$\frac{750 \times 452.4}{62.7} = 5400 \text{ ft. per min.}$$

The exhaust pipe is 10 in. in diameter and the speed of the exhaust steam is only 4300 ft. per min. With a receiver at the engine having a volume of 25 cu. ft., the size of the steam pipe might be reduced to 7 in.

Professor Barr's investigations (see Chapter XIV.) in 1896-97 showed an average velocity of 6500 ft. per minute for live steam and 4400 for exhaust steam in high-speed engines. The corresponding figures for low-speed engines were 6000 and 3800 ft. per minute, respectively.

137. Condensation in Piping.—The condensation in steam piping is due largely to radiation and conduction and not to any fall of pressure. This condensation with bare pipe is enormous and such as to practically forbid its use for carrying steam any considerable distance. The experiments most frequently quoted in this connection are those of Mr. George M. Brill in 1895* and those of Mr. George H. Barrus in 1902.†

The first series of tests were made on lengths of 60 ft. of 8-inch standard wrought-iron pipe, bare and covered with different

* Transactions A. S. M. E., Vol. XVII.
† Transactions A. S. M. E., Vol. XXIII.

kinds of commercial coverings. The following table is an abstract of those given in the report of the test:

TABLE II.

BRILL'S TEST OF PIPE COVERINGS.

MATERIAL.	Thickness inches.	Loss of Heat per hour per sq. ft. surface		Cost of 60 ft. of covering
		Lb. steam.	B.t.u.	
Bare Pipe		.8458	736.5	
Magnesia	1.25	.1200	104.5	$33.87
Rock Wool	1.60	.0802	69.8	29.62
Mineral Wool	1.30	.0890	77.5	23.34
Fire Felt	1.30	.1570	136.7	31.80
Manville Sectional	1.70	.1093	95.2	32.56
Manville Sectional and Hair Felt	2.40	.0662	57.7	40.56
Manville Cement	2.20	.1078	93.8	42.56
Champion Mineral Wool	1.44	.0990	86.2	18.87
Hair Felt	.82	.1319	114.9	13.50
Riley Cement	.75	.2979	259.4	28.71
Fossil Meal	.75	.2747	239.2	34.20

The prices are given merely to show the relative costs of the various coverings tried. From these values Mr. Brill figures an average net saving of over one dollar per square foot of pipe per year for all the coverings except the last two.

In 1901 Mr. George H. Barrus made a series of tests on 2-in. and 10-in. pipe at pressures of 80 lb. and of 150 lb. per square inch to determine the efficiency of various pipe coverings then on the market.

The condensation was effected on "dead ends" where the only circulation was that caused by the removal of the condensed steam. Experiments showed, however, that there was no appreciable difference in the rate of condensation when the steam was allowed to circulate.

The 2-in. pipes were each about 100 ft. long and the 10-in. pipes each about 35 ft. long. The following tables give a summary of the results:

TABLE III.

BARRUS' TESTS OF PIPE COVERINGS.

	Name of Covering.	Cost (applied) per running foot. Cents.	Diff. of Temp. between Steam and air. Deg.	Net surface Bare Pipe sq. ft.	Net condensation per hour six hours. Lb.	B.t.u. lost per deg. per sq. ft per hour.
	80 Lb. Section 2-in. Pipe.					
1.	Asbestocell	13.60	263.0	63.68	13.47	.728
2.	New York Air Cell	16.32	256.6	63.24	13.43	.750
3.	Carey's Molded	12.64	261.9	64.12	14.28	.768
4.	Asbesto-Sponge Molded	12.64	261.9	63.84	14.35	.778
5.	Gast's Air Cell	14.56	262.6	63.64	14.68	.793
	150 Lb. Section 2-in. Pipe.					
6.	A-S Hair Felt, 3-ply plain	23.89	303.5	64.41	10.22	.462
7.	A-S Hair Felt, 2-ply corrugated	20.11	304.9	64.21	10.86	.490
8.	A-S Felt, 59 laminations	23.20	304.1	63.78	10.76	.490
9.	A-S Felt, 48 laminations	23.20	294.0	64.21	11.35	.531
10.	Magnesia	25.12	300.6	63.66	11.50	.531
11.	Asbestos, Navy Brand	22.24	300.6	63.82	13.16	.606
	150 Lb. Section 10-in. Pipe.					
12.	A-S Felt, 76 laminations	78.30	302.0	97.10	9.29	.280
13.	A-S Felt, 66 laminations	59.00	315.0	97.10	10.60	.306
14.	Magnesia	71.00	299.2	97.10	11.64	.354
15.	Asbestos, Navy Brand	67.70	298.4	97.81	12.79	.387
16.	Watson's Imperial 1-in	41.00	303.0	97.81	14.37	.428
	Bare Pipes.					
17.	2-in. Pipes, 80 lb. pressure		273.2	63.70	59.16	3.081
18.	2-in. Pipes, 150 lb. pressure		305.2	63.98	74.40	3.366
19.	10-in. Pipes, 150 lb. pressure		295.4	100.45	107.84	3.220

A study of the results of these experiments show the relatively enormous loss from the use of bare pipe, and the comparatively small difference in efficiency of the various coverings tested. In other words, any pipe covering is a great improvement, and of those in common use, one is not very much better than another.

Mr. Barrus' tests also show that the loss per square foot of surface is greater in small pipes, as would be expected.

138. Measurement of Moisture in Steam.—Before testing a sample of steam to determine the quantity of water entrained, it is necessary to so collect the sample that it may fairly represent the steam in the pipe. This is a matter of considerable difficulty, since the entrained water is not uniformly distributed. In horizontal pipes, some of the water runs along the bottom of the pipe, while the water descending with the steam in a vertical pipe may carom from side to side. Probably, the best location for collecting samples is in a vertical pipe carrying an upward current of steam.

The fact has been pretty thoroughly demonstrated that steam which is slightly superheated and moving with considerable velocity may carry along drops of water for some distance. The steam being a poor conductor, if there are no convection currents, the evaporation of the water will take time.

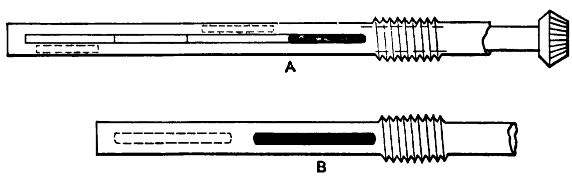


Fig. 146. Sampling Tubes.

139. Sampling Tubes.—The usual form of tube is a half inch brass or iron pipe extending nearly across the steam pipe, open at the inner end and perforated transversely with several small holes. The quality of the sample will depend somewhat upon the location of these holes. If we consider some or all of the water to consist of minute drops or streams moving along with great velocity, it is evident that the inertia of the water will tend to prevent it from entering holes in the sample tube, unless they face the current. Furthermore, if such holes extend entirely through the tube, the drops of water might pass through and not appear in the sample. In other words, the tube itself may act as a separator and thus defeat the object of the test.

A tube used by Prof. C. C. Thomas in connection with his electric calorimeter overcomes this difficulty. (See *A*, Fig. 146.) A

brass tube, extending a little over half way across the steam pipe, has a longitudinal slot along its whole length on the side facing the steam current. Inside this is another brass tube capable of rotation in the first. The inner tube has a series of short slots arranged spirally along its surface, so that only one at a time is brought to coincide with the outer slot. The inner tube is revolved by bevel gears and a hand wheel at its outer end. When the hand wheel is turned to position 1, the slot is open next the side of the steam pipe. When turned to position 2, the opening is a little nearer the center; when turned to the last position, the opening is at the center of the steam pipe.

Samples can thus be taken from any part of the pipe at will, by turning the hand wheel, and are always taken on the side of the tube facing the steam current.

For small steam pipes from 3 to 6 in. in diameter, the author has used a single tube with the inner end closed and with two slots 180 deg. apart, one near the side of the steam pipe and one at the center. By rotating the tube 180 deg. in the pipe samples can be taken from these two points without much trouble. (See *B*, Fig. 146.)

140. The Barrel Calorimeter.—The method formerly in use for moisture determinations consisted in running a known weight of the steam to be tested into a barrel containing a definite quantity of cold water, and then, measuring the rise of temperature of the water caused by the condensation of the steam.

The statement of the problem would be as follows:—

Let $x =$ quality of steam.
$L =$ its latent heat.
$q =$ liquid heat.
$w =$ weight used.
$W =$ weight of cold water.
$t_1 =$ initial temperature of water.
$t_2 =$ final temperature of water.

The heat given up by the steam $= w(xL + q - t_2 + 32)$, and that received by the water $= W(t_2 - t_1)$. Neglecting heat lost by conduction and radiation, these two are equal.

Equating and solving

$$x = \frac{W(t_2 - t_1) - w(q - t_2 + 32)}{wL} \qquad (49)$$

The chance of error with this apparatus is considerable, and it cannot be relied upon for accurate determinations.

The precautions to be taken are:

(1) A warming up of the barrel before any experiment is made.

(2) Thorough mixing of the water in the barrel either by the steam jet itself or by mechanical means.

(3) Great care in weighing the water.

Example.—Let the pressure of the steam be 80 lb. by gage. Let the initial and final temperatures of the water be 54 and 112 deg. Fahr. and the corresponding weights be 240 and 254 lb. Required the quality of the steam in the pipe.

From the steam tables we find $L = 886.7$

$$q = 294$$

Substituting values in the formula we have:

$$x = \frac{(240 \times 58) - 14\,(294 - 112 + 32)}{14 \times 886.7} = \frac{13{,}920 - 2996}{12{,}414} = 0.88$$

This method cannot be recommended except for rough determinations where there is much water present. Errors in weighing and in getting the exact final temperature lead to variations of from 1 to 2 per cent in the results.

141. The Separating Calorimeter.—A calorimeter has been devised by Prof. R. C. Carpenter which mechanically separates the moisture from the steam. The construction of the apparatus is shown in Fig. 147. The steam from the sampling tube enters the calorimeter through the pipe *A*, and is discharged downwards into the cup *B*. The course of the steam and water is here reversed, with the result that the water is thrown outward through perforations in the cup and collects in the inner chamber *C*, where it is measured by the gage glass *D*. The steam passes upward and then downward into the outer chamber, whence it escapes through

a standard orifice *E* into the air. The apparatus is thus jacketed by the escaping steam, which is maintained at a high pressure by the throttling at *E*. A gage at *G* shows the pressure of the steam and the corresponding discharge in pounds per 10 minutes. The calculations with this instrument are very simple and tests show the steam discharged from it to be practically dry.

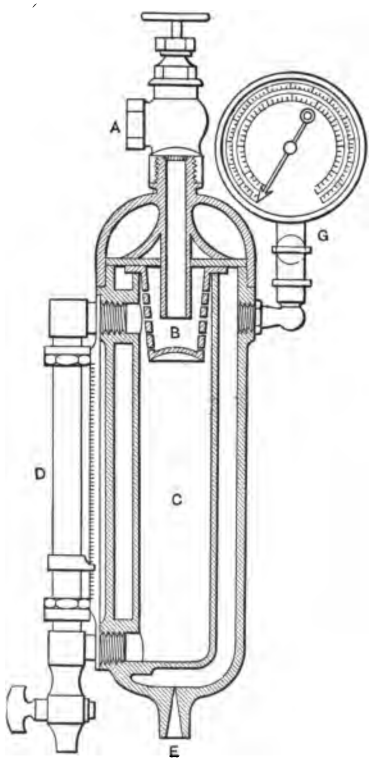


Fig. 147. Separating Calorimeter.

142. The Throttling Calorimeter.—Prof. C. H. Peabody designed in 1888 a calorimeter in which the moist steam was dried and then superheated by throttling, the degree of superheat depending on the initial condition of the steam. Fig. 148 shows this instrument as usually constructed. The steam enters the

calorimeter through the small orifice at A, its pressure being reduced nearly to that of the atmosphere.

The heat due to this difference of pressure is used in evaporating the moisture in the steam and in superheating the steam in the calorimeter. Or, to express it in another way, by referring to Fig. 145, the heat shown by the shaded area is not converted into kinetic energy as in the ordinary nozzle or orifice, but used up in moving

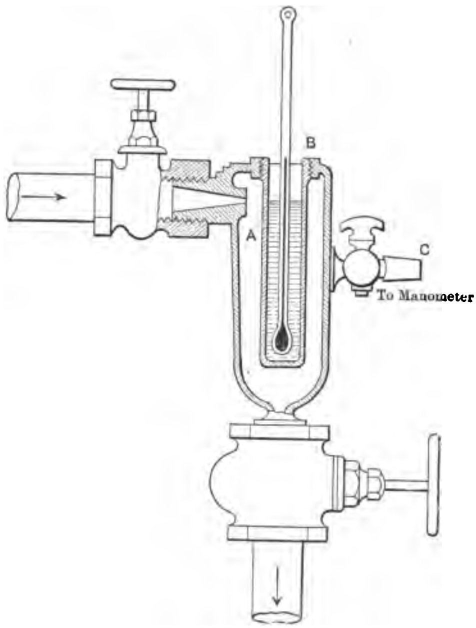


Fig. 148. Throttling Calorimeter.

the point D further to the right until it reaches or passes the line of saturation EF. The velocity of the steam in the calorimeter is so slight that it may be neglected. The instrument and its connections to the pipe should be protected from radiation.

A thermometer inserted in the cup at B shows the final temperature and a mercury manometer is attached at C to indicate the pressure. It is also necessary to read the barometer during the

test. It is then assumed that the heat in the steam is the same before and after entering the calorimeter.

Let p = pressure of steam in pipe.
L = corresponding heat of evaporation.
q = corresponding heat of liquid.
x = quality of steam in pipe.
p_1 = pressure inside calorimeter.
t_1 = corresponding temperature.
H_1 = total heat of steam at t_1.
t_2 = actual temperature in calorimeter.

The heat in the steam before throttling is then:

$$xL + q,$$

and after throttling:

$$H_1 + 0.5(t_2 - t_1)$$

(the value 0.5 being assumed correct for atmospheric pressure and low superheat).

Equating the two heats and solving:

$$x = \frac{H_1 - q + 0.5(t_2 - t_1)}{L}$$

A chart or diagram can be prepared from this formula, which is convenient when much of this work is to be done.

It is evident that only a certain amount of moisture can be evaporated in this way.

Putting $t_2 - t_1 = 0$ in the equation we have:

$$x = \frac{H_1 - q}{L}$$

as the limiting value of x which can be determined by throttling. Since p_1 is usually about 15 lb., we may put $H_1 = 1147$. The limiting values of x for several pressures will be:

Absolute pressure	q	L	x
75	277	899	96.8
100	298	884	96.
125	315	872	95.5
150	330	861	94.8

The throttling calorimeter is generally regarded as an accurate instrument within the limits just indicated. If it is well protected from radiation, it should give reliable results. In view of what

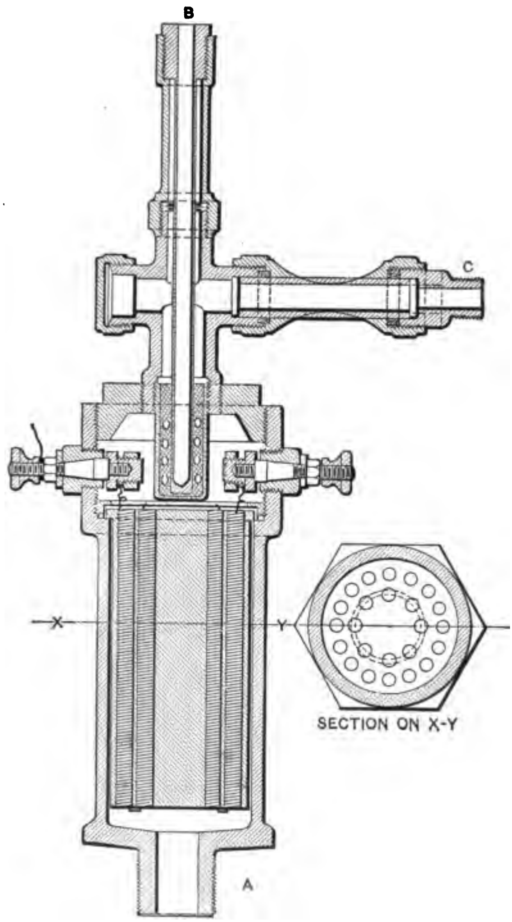


Fig. 149. Electric Calorimeter.

has been said concerning water carried along by saturated steam, there is a possibility that the thermometer in the calorimeter may show a higher degree of superheat than the average of the mix-

ture. The introduction of layers of fine wire gauze between the orifice and the thermometer cup has been suggested as a means of mixing the steam thoroughly.

Example.—The pressure in a steam pipe is 130 lb. gage; the pressure in the calorimeter is 1.4 in. of mercury and the temperature is 242 Fahr. The barometer pressure is 28.9 in.

Results:

Atmospheric pressure	=	14.2 lb.
Pressure in pipe	=	144.2 lb.
Pressure in calorimeter	=	14.9 lb.

$$x = \frac{1146.8-326.7+0.5\,(242-212.6)}{863.5} = 0.966$$

143. Electric Calorimeter.—A calorimeter recently designed and put upon the market by Prof. C. C. Thomas is the most satisfactory instrument yet devised for determining the moisture in steam.

Briefly stated, its operation consists in causing steam to flow at a regular rate through the instrument and drying it en route by the heat from electrical resistance coils.

The steam enters at *A*, Fig. 149, passes up through small circular conduits containing coils of fine wire heated by an electric current, and out through a glass observation tube *C*. A thermometer inserted at *B* shows the temperature after drying. *E* and *F* are the electrical terminals.

The calorimeter is connected up in series with a water rheostat and a wattmeter and an electric current of about 125 volts is used. A constant flow of steam is established by means of a regulating valve. As long as the steam is saturated, the thermometer at *B* shows a constant temperature and water can be seen in the observation tube. As soon as sufficient current has been supplied to dry the steam, the thermometer shows a sudden rise of temperature and the steam in *C* clears up. A reading of the wattmeter shows the energy being used, which is called E_1.

The steam is then superheated an amount *S*, as shown by the thermometer, and a new reading taken of the wattmeter, called E_2.

The weight of steam passing can then be calculated from the energy required to superheat it and from this can be determined

the heat H_x added to each pound of steam to dry it. Let H_v be the heat of vaporization of a pound of dry steam, then will the quality

$$x = \frac{H_v - H_x}{H_v}$$

As shown by the inventor*, $H_x = K \frac{E_1}{E_2}$ where K is the constant of the calorimeter and can be determined from a chart for different degrees of superheat.

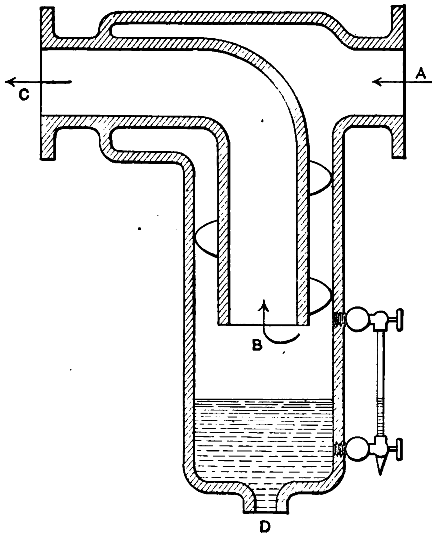


Fig. 150. Steam Separator.

This is the instrument used by Professor Thomas, in his investigations on the specific heat of superheated steam before alluded to.

It will be readily seen that by measuring the flow of steam with a condenser and at the same time determining the energy required for superheating, it is possible to measure the specific heat directly.

144. Separators.—A separator is a cast-iron receptacle through which the steam passes on its way to the engine, and which removes mechanically a part of the entrained water. This

*"Power," November, 1907.

water is removed sometimes by baffles which deflect the stream and sometimes by centrifugal action. In general, advantage is taken of the fact that the greater density of the water causes it to leave the steam path whenever that path curves from a direct line.

Fig. 150 illustrates a separator embodying this principle. The steam and water entering at *A* pass downwards in a spiral path through the annular passage into the tank at *B*. Here the direction of the steam is reversed ,and it passes out at *C*. The water is separated from the steam by gravity and by centrifugal force

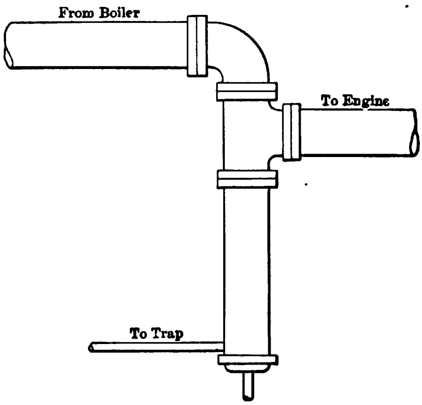


Fig. 151. Steam Pipe with Drop Leg.

and collects in the tank at *B,* whence it is removed by a pipe and steam trap through the orifice at *D*.

Such a separator will remove the bulk of the water from wet steam, leaving perhaps from one half to one per cent to go through the engine. As the capacity of the separator and its drain trap is limited, it is always well to have a drop leg on the main pipe to prevent damage to the engine from any sudden overflow of water, as shown in Fig. 151. At the point where the boiler header drops down to the level of the steam main in the engine room, a drop leg is connected, which may extend to the floor if desired, and which is drained by a pipe and steam trap.

145. Drainage of Piping.—The general design of steam piping will be considered in another place, but the rules for avoiding trouble from condensation should be noticed in this connection. All steam mains should be so sloped that the water will run in a direction away from the boiler and thus allow gravity and the flow of steam to work together. The water should be removed by "bleeders" or drain pipes at the ends of the various steam mains and trapped back to the hot well. No low places, where water can collect, should be allowed, but if they are unavoidable, they should be drained as described above.

Steam pipes to all engines and auxiliaries should rise from the top of the main and bend over and down, so as to prevent any water from getting to the engine. Even then, it will be necessary to provide a drain pipe above the throttle to remove water which accumulates when the engine is not running.

146. Lubrication.—Two methods are in common use for supplying oil to the interior of the cylinders of a steam engine, gravity feed and the oil pump. For the smaller sizes of engines and for auxiliaries such as feed and air pumps, the gravity feed is generally used. Fig. 152 shows the principle of the ordinary sight feed cylinder lubricator as applied to the steam pipe of an engine.

The metal chamber *C* contains oil in the upper part and water in the lower part. When the valve at *E* is opened, steam condenses in the small pipe and in the bulb *G* and a column of water forms above the lubricator. The valve at *F* is then opened, and the weight of the water column forces the oil into the pipe, since the steam pressure is the same at *E* and *F*. Water is slowly forced down through the tube *H* and oil out through the tube *I*. The valve at *K* regulates the flow of the oil and compels the latter to escape in drops which rise through the water in the tube *M*. The other glass tube *N* shows the level of the water *AB* in the lubricator, and the plug at *D* is used when refilling the chamber with oil.

On large engines, the cylinder oil is usually fed in by a small force pump driven by the engine and communicating with the various cylinders through brass or copper tubes. Glass sight feeds

may be used on these pipes, or a tell-tale valve may be opened to show when the pump is working properly.

For lubricating the various bearings and slides outside the cylinder, a gravity system is usually employed. The oil is contained in a tank some distance above the engine and the oil distributed by pipes to the various sight-feed oilers located at the different bearings.

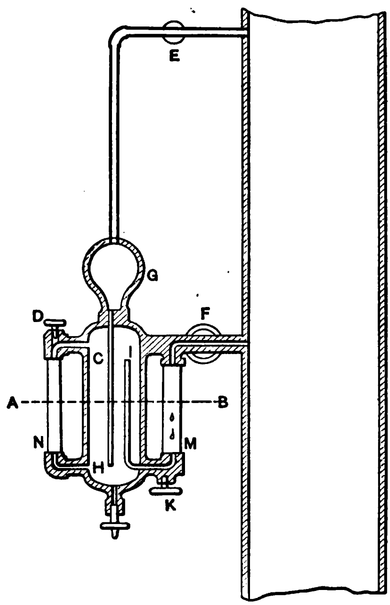


Fig. 152. Sight-Feed Lubricator.

Another principle which must be kept in mind is that every oiler must be stationary and the oil conveyed to the moving parts by tubes, wipers, etc., in such a way that all of the cups can be kept in order without stopping the engine. The collection of surplus oil and its return to the system after filtering is an important factor in engine economy. Frequently, the main bearings of large engines are fitted with independent oil pumps and a continuous stream of oil is poured over the journal. Some details of lubricating devices will be considered in Chapter XIV.

PROBLEMS.

1. Find the velocity of steam flowing through an orifice into the atmosphere from a boiler where the pressure is 60 lb. by gage; when it is 80 lb. by gage. Assume dry steam in the boiler.

2. Find the weight of steam discharged per minute in Problem 1, if the orifice is circular and 0.1 inch in diameter.

3. Solve Problem 2 by Napier's formula.

4. A steam main is 360 ft. long and 8 in. in diameter. Find the weight of steam that can be carried by this main at 150 lb. initial pressure with a loss of pressure not to exceed 5 lb.

5. Determine the probable fall of pressure in a steam pipe 6 in. in diameter and 120 ft. long with two elbows, if the flow of steam is 7200 lb. per minute with an initial pressure of 120 lb. per square inch.

6. Calculate the size of standard steam pipe for an engine of 250 i.h.p., using 20 lb. of steam per horse power hour at 105 lb. pressure and with cut-off at ¼ stroke.

7. Make a similar calculation for an engine having a cylinder 20 by 30 in. and a speed of 120 r.p.m.

8. Determine the probable condensation of steam in pounds per hour for the pipe in Problem 4, if the metal is bare. If covered with magnesia.

9. In using a barrel calorimeter, the initial and final weights of water were found to be 380 and 402 lb. and the temperature 62 and 117 deg. Fahr. If the initial pressure of the steam was 110 lb., what was the probable moisture?

10. Observations with a throttling calorimeter showed a temperature of 262 deg. in the calorimeter and a pressure of 0.8 in. of mercury. The steam pressure in the pipe was 114 lb. by gage and the barometer showed 29.6 in. Required quality of the steam in the pipe.

CHAPTER XIII.

STEAM ENGINE PERFORMANCE.

147. In General.—The work done by an engine is measured by its horse power, as shown by the indicator or by the brake. The energy supplied by the boiler is measured by the quality and quantity of steam used per unit of time.

The economy, or, as it is frequently called, the performance of an engine is therefore measured by the ratio of the two quantities mentioned above. Engine performance is usually expressed in terms of the steam used per indicated horse power per hour. Since the quality of the steam may vary on account of difference in pressure and temperature, this is not an exact method of state-

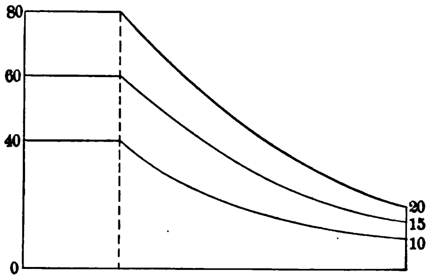


Fig. 153. Diagrams from Throttling Engine.

ment. The performance of any heat engine should be expressed in heat units per horse power hour, or in heat units per foot pound. This point will be discussed more fully in another place. (See Art. 155.)

148. Willans' Law.—It was first shown by P. W. Willans that the steam consumption per hour is theoretically proportional to the horse power, if the ratio of expansion remains the same; i.e. if the engine is throttled to reduce the power. This is best shown by a diagram, as in Fig. 153. Let steam be expanded four times in each case and let the initial pressures be 80, 60 and 40 lb. in the three diagrams shown. Assuming hyperbolic expansion,

the terminal pressures will be 20, 15 and 10 lb. respectively and the mean pressures, as calculated from the formula,

$$p = p_2 \left(1 + \log_e \frac{v_2}{v_1} \right)$$

would be proportional to the terminal pressures and equal to $2.386p_2$ or $p = 47.7$, 35.8, 23.9 lb., respectively.

Since the steam at the end of expansion occupies the same volume in each case and since the density is proportional to the pressure, the weight of steam used by the engine will be proportional to the terminal pressure. The power developed by the engine is measured by the mean pressure, and as this has been

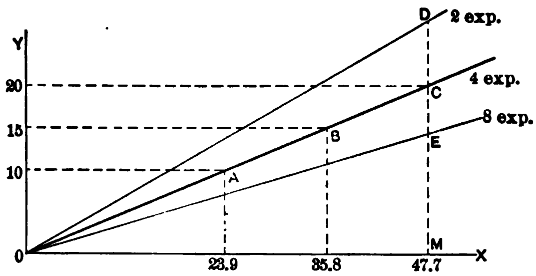


Fig. 154. Willans Straight Line Diagram.

shown to vary as the terminal pressure, the power developed will be proportional to the terminal pressure, and therefore, to the weight of steam used. This can be shown graphically by the so-called Willans straight line diagram, Fig. 154. The mean pressures are laid off on the axis OX and the terminal pressures on OY to the same scale. Using these distances as coördinates, the points A, B and C are located and will lie in one straight line passing through O. From what has gone before, it is evident that the tangent of the angle COY is

$$\left(1 + \log_e \frac{v_2}{v_1} \right)$$

or 2.386 in this instance.

Similarly, with two expansions, the terminal pressures will be 40, 30 and 20 lb., and the mean pressures will be $1.693p_2$ or 67.7, 50.8 and 33.9 lb. The line *OD* on the diagram represents these values.

For eight expansions, the value of the ratio is $(1 + \log_e 8) = 3.079$, as shown by the line *OE* in Fig. 154. The relative economies for any particular mean pressure, as for instance 47.7 lb., are shown by comparing the distance *ME, MC* and *MD* on that ordinate, since these distances indicate the relative water-rates.

149. Variable Cut-off.—The effect of varying the cut-off instead of the initial pressure, i.e. using an automatic cut-off governor instead of a throttling governor, is shown in a similar

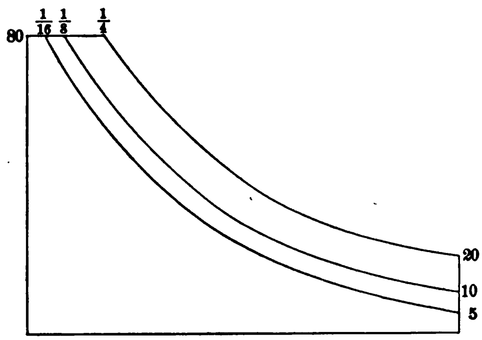


Fig. 155. Diagrams Showing Variable Cut-off.

manner. Starting with the same diagram as in Fig. 153, a reduction of power is accomplished by increasing the number of expansions to 8 and to 16, the initial pressure remaining the same, Fig. 155. The terminal pressures will be accordingly 20, 10 and 5 lb., and the weight of steam used will vary accordingly. The mean effective pressures will be 47.7, 30.8 and 18.8 lb.

Plotting the points, as in Fig. 154, gives the curve *ABC* in Fig. 156. As this line is entirely beneath the line *OC,* it is evident that the theoretical economy is improved by varying the cut-off to reduce the power. If the engine in question had been regulated by throttling from the condition *C,* when the mean pressure had

been reduced to 30.8 lb., the steam consumption would have been represented by *ND* instead of *NB*. The line *CBA* produced would intersect the line *CO* at the origin, since in any ideal engine the steam consumption is zero for no load.

150. Actual Performance.—The actual steam consumption of an engine, as determined by the weight of condensed steam and the indicated horse power, varies widely from that shown by the foregoing diagrams. The reasons for this have been noticed in preceding chapters and may be summarized as follows:

1. The friction of the engine.
2. Condensation in the cylinder.
3. Valve and piston leakage.
4. Loss by radiation.
5. Jacketing, superheating, etc.

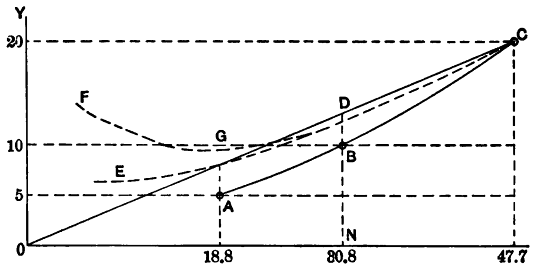


Fig. 156. Showing Effect of Cylinder Condensation and Leakage.

The friction of the engine is nearly constant at different loads, as has been repeatedly proved by experiment. A slight difference in the lubrication, in the tightness of the stuffing boxes or the tension of the piston packing will make much more difference than a change of load. This constant load due to friction reduces the efficiency particularly at light loads, so that the steam line, Fig. 156, cuts the axis of *Y* above the origin, as shown by the dotted line *CE*. The effect of the various losses numbered (2), (3) and (4) is to change the shape of the indicator diagram, and in general, to reduce its area and mean pressure. The remedies included

under (5) partially off-set the losses before mentioned, as has been previously explained.

The losses caused by cylinder condensation and leakage are greater at light loads, when the power is regulated by varying the cut-off, since there is a wider range of temperature and pressure. (See Chapter X.) This tends to raise the left end of the curve in Fig. 156, giving it somewhat the appearance of the line *CGF*.

Since the terminal pressures in a real diagram may not be proportional to the steam consumption, it will be better to plot subsequent curves in terms of the actual steam consumption in pounds per hour and the indicated horse power. The ratio of any ordinate to its abscissa will then give the water rate for that particu-

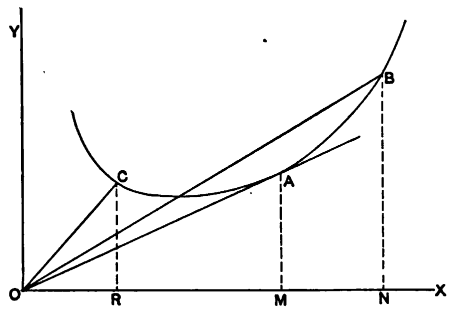


Fig. 157. Showing Construction of Water-Rate Diagram.

lar load. This ratio is the tangent of the angle between the axis of *X* and a line drawn from *O* to the point in question. If a line be drawn from *O* and tangent to the curve, as at *A*, Fig. 157, $\frac{AM}{OM}$ is less than for any other points on the curve as *B* or *C*. *OM* then represents the indicated horse power at which the water rate is lowest. To the right of *A*, incomplete expansion is the cause of the poorer efficiency, while to the left of *A*, increased condensation and leakage are responsible. If the curve is nearly straight at *A* and departs slowly from the straight line, the engine is efficient with varying load. The endeavor should then be to make the

angle *AOM* as small as possible and keep the line *BAC* approximately straight within the limits of load variation.

The following are some of the results obtained in testing a 16 by 16, non-condensing four-valve engine:

Indicated horse power.	Weight of steam per hour.
220	5190
176	3990
132	2930
89.5	2140
45.7	1385
(Friction) 8.6	810

In Fig. 158, the horse powers are taken as abscissas and the weights of steam as ordinates, and the curve *CAB* is drawn cor-

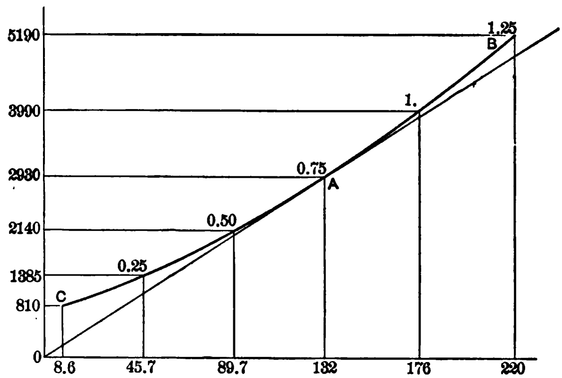


Fig. 158. Water Rate at Different Loads, 16 x 16 Engine.

responding to the curve *CAB* in Fig. 157. The numerals at the curve points refer to the fraction of the normal load, this engine being rated at 175 horse power. The line *OA* being drawn tangent to the curve touches it at *A*, showing the engine to be most economical at about three quarter load. The curve is nearly straight in this vicinity, and the efficiency is therefore nearly con-

stant between one half and full load. The water rates for these three points are as follows:

0.50 load	23.89
0.75 "	22.24
1.00 "	22.74

The point *C* shows the friction load, or the power required to run the engine itself. The ratio of the friction to the normal load is only 4.88 per cent. The straightness of the line *CAB* and the low value of the friction are both evidences of an unusually good economy.

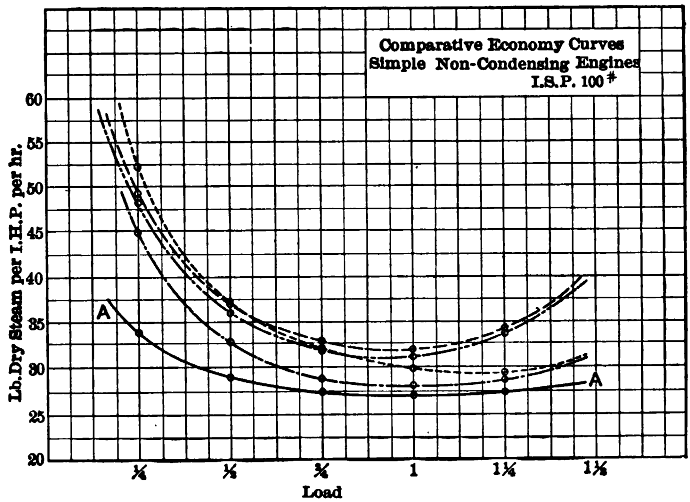


Fig. 159. Comparative Economy Curves.

151. Water Rate Curves.—Instead of plotting the total weight of steam used, as in the preceding figures, most engine builders prefer to use the weights of steam per horse power per hour, or the "water rates," for ordinates to the curve, and to take the boiler pressures, the mean effective pressures, or the ratios of expansion as abscissas. This curve has the advantage of showing to the eye the minimum water rate and the effect of varying the pressure or number of expansions. Fig. 159 shows the economy

curves of five simple expansion engines plotted from the results of tests. In this figure, the mean effective pressures expressed in terms of the rated or normal load, are used for abscissas.

A study of these curves shows: (1) A minimum water rate at or near full load. (2) A rapid increase in this rate for loads less than one half the normal. (3) A more gradual increase for overloads. Curve *A* not only shows the least steam consumption at full load, but is straighter than the others, indicating a wider range of economical performance.

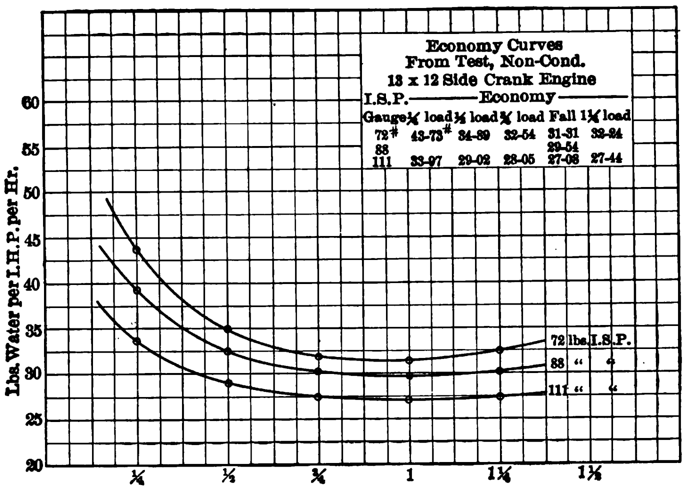


Fig. 160. Economy Curves.

In Fig. 160 are given three curves from a simple non-condensing 13 by 12 engine for different initial pressures, as plotted from the results of tests. This diagram indicates clearly the advantage of a high initial pressure in an engine of this class, since the lower lines on the chart not only show less steam used, but are straighter than the upper one.

152. Performance of Compound Engines.—The reasons for the better economy of a compound engine have been explained in Chapter VII. Some examples of actual performance will be

given here and the results compared. The curve in Fig 161 is that of a tandem compound, condensing engine, 13 and 23 by 15 in. with piston valves. The engine was running at about 250 r.p.m. with about 21 in. vacuum. This engine shows good economy under a varying load which is not always the case with compounds.

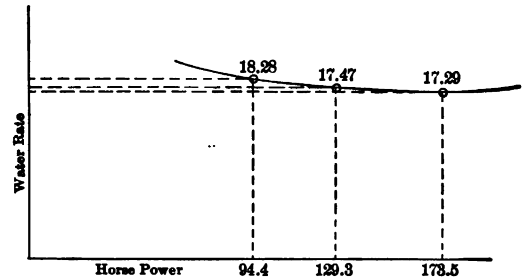


Fig. 161. Water Rate, 13 and 23 by 15 in. Engine.

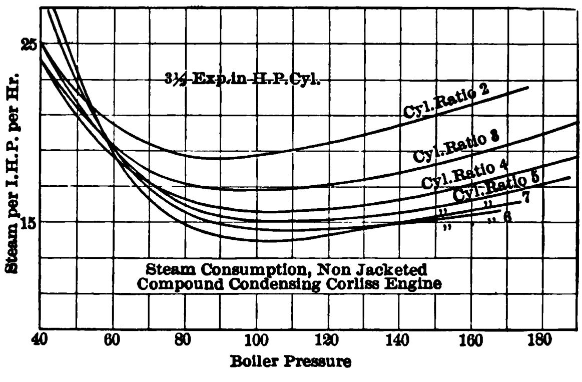


Fig. 162. Economy Curves, Corliss Engine.

The effect on the steam distribution of varying the cut-off in two cylinders or of changing the initial pressure in the high cylinder has been discussed in Chapter VII. In general, high expansion of the steam in a two-cylinder compound can be obtained by an early cut-off in the high pressure cylinder or by a large ratio between the two cylinders.

In Figs. 162, 163 and 164 are shown economy curves from non-jacketed, compound condensing Corliss engines. These curves were plotted from the results of a large number of tests on this type of engine, and were furnished by a well-known firm of engine builders.

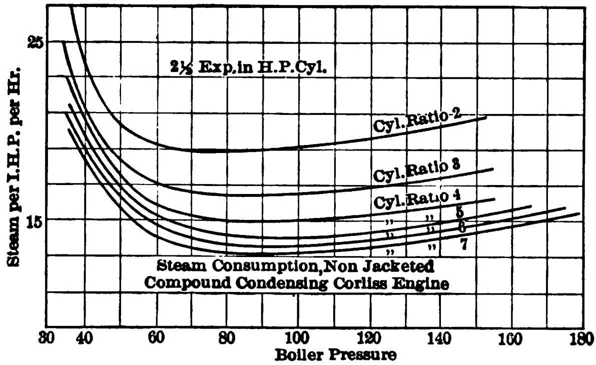


Fig. 163. Economy Curves, Corliss Engine.

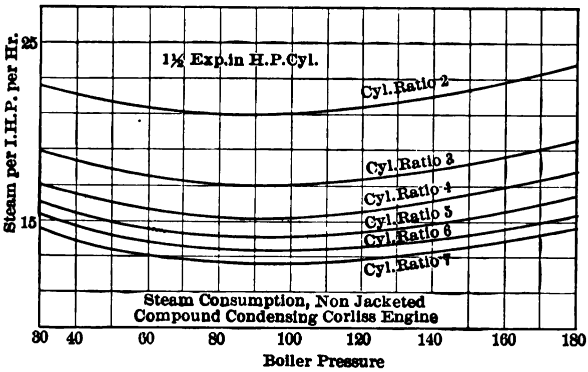


Fig. 164. Economy Curves, Corliss Engine.

The following conclusions can be drawn from these figures:

(1) A late cut-off in the high-pressure cylinder combined with a large cylinder ratio is the most economical arrangement.

(2) Such an arrangement not only gives the lowest water rate,

but it also gives a straighter line or a more uniform rate for different initial pressures.

If one and a half expansions are used in the high-pressure cylinder and a cylinder ratio of seven, the boiler pressure may be changed from 130 to 60 lb. without any noticeable fluctuation of the water rate. The curves shown in Fig. 162 would not allow of any such variation.

Such engine builders as have been consulted by the author agree in saying that greater economy is effected by throttling the steam in the high pressure cylinder for light loads than by any adjustment of the cut-off, but data are lacking as to the amount of saving which can be obtained in this way.

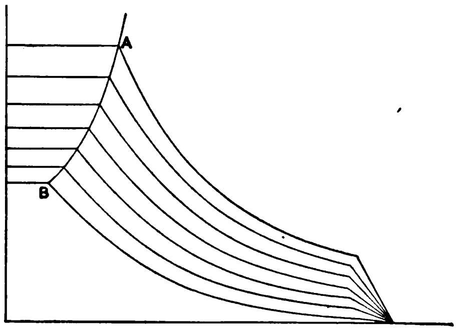


Fig. 165. Variation in Initial Pressure and Point of Cut-off for best Economy.

As has been pointed out in a discussion by Mr. E. J. Armstrong,* there are certain initial pressures and rates of expansion which will give the best economy in any particular engine, for various loads, and these will have to be determined by experiment for each class of engine. Mr. Armstrong illustrates this by a series of superimposed diagrams as in Fig. 165. The points of cut-off will then lie along some curve as *AB* such that both pressure and expansion vary.

* Transactions A. S. M. E., Vol. XVIII.

153. Condensation and Leakage.—The losses due to initial condensation of steam in the cylinder and to leaking of steam past the piston and valves have already been discussed in Chapter X. The report of Professor Capper of London to the Steam Engine Research Committee of the British Institution of Mechanical Engineers, in 1905, treats in part of these two losses and shows results of what is probably the first attempt to separate them from each other. The experiments of Professor Capper showed the leakage past the piston to be independent of the revolutions per

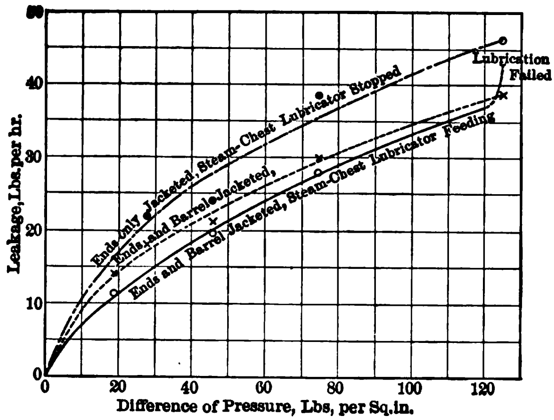


Fig. 166. Variation in Slide Valve Leakage with Pressure.

minute and to be proportional to the pressure of admission and the area of the diagram. At its worst, this leakage amounted to less than 2 per cent of the steam used by the engine. On the other hand, the leakage past the slide valve employed in these experiments was proved to be considerable, varying from 4 to 20 per cent under different conditions.

This leakage increases as the difference of pressure on the two sides of the valve; when the valve is stationary, this increase is approximately proportional to the difference in pressure, but when the engine is running, the rate of increase is less, and decreases with the increase of speed. Thorough lubrication of the

valve tends to diminish the leakage, and the use of a steam jacket on the cylinder has the same effect.

From these facts Professor Capper concludes that much of the leakage is caused by a condensation of steam on the valve face and a subsequent reëvaporation into the exhaust cavity. The same remedies which can be relied upon to reduce initial condensation will be effective in preventing leakage.

Fig. 166 shows the variation of slide-valve leakage with pressure and also the effect of steam jacketing and lubrication, as determined by Professor Capper.

In the same year Professor Mellanby of Manchester reported to the British Institution the results of his experiments on steam jacketed compound engines. In this report, there is evidence that a great part of the loss of heat in the engine cylinder is due to leakage; the effect of jacketing is to reduce this as well as the cylinder condensation. Anything which tends to keep the entering steam dry will have a beneficial effect as regards both of these losses.

154. Performance of Non-condensing Engines.—In a paper read before the American Society of Mechanical Engineers,* Mr. J. B. Stanwood has calculated the theoretical water rates for the Rankine cycle at various pressures and has compared with these the actual water rates of various engines as determined by efficiency tests. Tables IV. and V. are reproduced from his paper. By cylinder efficiency he means the ratio of the theoretical to the actual water rate. This is resolved into two factors, one the "Condensation factor" C which covers the losses due to condensation and leakage and the other the "expansion factor" F which shows the loss caused by incomplete expansion and the drop in pressure at the end of the stroke.

An early cut-off and consequent high expansion ratio will increase F, but will usually diminish C. For each particular engine and each initial pressure, there is a certain ratio of expansion which will give to the product of these two factors, $FC = E$, a maximum value, E being the ratio of the area of the actual indicator diagram to that of the Rankine cycle having the same range of pressures and the same quantity of steam.

*Transactions A. S. M. E., Vol. XXVI.

TABLE IV.

Boiler Pressure by Gage.	Available B.t.u.	Minimum Theoretical Water Rate.	Corresponding Theoretical m.e.p.
P.	*U.*	*S.*	*M.*
60	117.0	21.8	26.1
70	126.3	20.2	28.3
80	133.6	19.1	30.3
90	141.0	18.1	32.0
100	147.3	17.3	33.6
110	152.3	16.7	35.1
120	157.6	16.2	36.5
130	162.6	15.7	37.8
140	167.4	15.3	39.0
150	171.7	14.8	40.1
160	175.6	14.5	41.2
170	179.3	14.2	42.3
180	183.2	13.9	43.3
190	186.8	13.6	44.2
200	190.2	13.4	45.2

Fig. 167 illustrates these comparisons. AB represents the volume of dry steam supplied to the engine in one stroke and BC is the adiabatic curve. Then $ABCD$ is the Rankine cycle for these

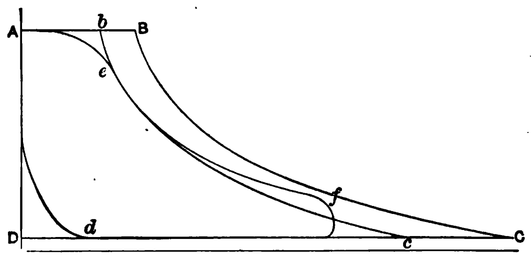


Fig. 167. Comparison of Rankine Cycle with Actual Diagram.

conditions. If $Aefd$ is the actual diagram, draw the adiabatic bc through e; then $AbcD$ is a Rankine cycle assuming the steam dry at cut-off. The ratio of areas $\frac{Aefd}{ABCD}$ gives E the cylinder efficiency, while the ratio of areas $\frac{Aefd}{AbcD}$ gives F the expansion

TABLE V.

Cylinder Efficiencies, Condensation and Expansion Losses in Non-condensing Steam Engines.

By Whom Tested.	Type.	Size.	r.p.m.	m.e.p.	Water Rate.	Steam Pipe Pressure by Gage.	Cut-off.	Cylinder Efficiency E.	Condensation Factor C.	Condensation Loss 1-C.	Expansion Factor F.	Expansion Loss 1-F.
		ins.				*lb.*						
Barrus, No. 23....	Single Valve, Automatic...........	8 x 12.........	303.7 307.8 304.5	24.5 34.8 43.1	31.78 31.20 30.63	83.0 82.4 81.9	.203 .312 .378	.590 .606 .611	.642 .720 .762	.358 .280 .238	.92 .84 .80	.08 .16 .20
Barrus, No. 1.....	Corliss, Simple.....................	23 x 60........	74.7	33.1	27.77	72.8	.367	.718	.84	.16	.85	.15
Barrus, No. 2.....	Corliss, Simple.....................	28½ x 59½....	64.8	41.2	25.8	101	.315	.671	.817	.183	.82	.18
Barrus, No. 31....	Corliss, Twin........................ * M.e.p.'s based on 85 r.p.m. E based on 100 lb. pressure.	2-16 x 42......	87.6 .... 84.9	5.0 *13.3 *19.4 *29.5 *38.1 48.4	78.60 38.40 31.40 25.80 25.40 25.91	101.8 .... 98.6	.041 .084 .121 .178 .231 .323	.236 .453 .552 .673 .685 .670	.382 .509 .588 .709 .745 .796	.618 .491 .412 .291 .255 .204	.620 .890 .940 .945 .920 .840	.380 .110 .060 .055 .080 .160
E. J. Armstrong..	Single Valve, Automatic........... E based on 100 lb. pressure.	13 x 12........	282 281 281 280 279 277	3.5 19.0 32.1 45.5 56.5 67.6	52.80 31.10 29.90 28.75 28.80 30.25	100	.052 .130 .210 .320 .420 .530	.333 .559 .582 .605 .604 .575	.300 .620 .660 .720 .775 .790	.672 .380 .340 .280 .225 .210	1.110 .901 .880 .840 .780 .730	.110 .099 .120 .160 .220 .270
Barrus, No. 20b...	4-Valve, Gridiron Type.............	28 x 60........	50.5	23.8	30.16	65.1	.222	.67	.722	.278	.92	.08
Barrus, No. 54....	Single Valve, Compound...........	12 x 20 x 13...	275.7 271.2 273.4	18.3 33.8 43.2	24.99 21.14 21.25	166.9 166.8 164.6	.221 .319 .520	.572 .676 .676	.686 .744 .814	.314 .256 .186	.834 .910 .830	.166 .090 .170
Barrus, No. 44....	Vertical Single Acting Compound.	11 x 19 x 11...	296.3	46.1	23.91	135.9	.605	.684	.803	.197	.85	.15
Barrus, No. 45e...	2-Valve, Riding Cut-off Compound.	14 x 23 x 24...	165.6	21.8	23.24	118	.487	.706	.794	.216	.89	.11
Barrus, No. 46....	4-Valve Compound............ {A / B}	17½ x 28 x 48.	69	22.15 34.40	22.11 21.59	128.7 135.5	.309 .419	.711 .712	.806 .816	.194 .184	.88 .87	.12 .13
P. W. Willans....	Willans' Vertical Compound.......	$9\frac{5}{16}$ x $13\frac{7}{16}$ x 6.	405	41.85	20.35	135.4	.393	.758	.883	.117	.86	.14

factor—a measure of the losses due to incomplete expansion, to release, compression and wire drawing.

The difference between the areas *AbcD* and *ABCD* measures the loss due to initial condensation and leakage. Mr. Stanwood shows that leakage past the admission valve, reëvaporation and the heat received from jackets and reheaters may increase the area of the diagram and the apparent efficiency, while at the same time they so increase the steam to be accounted for as to neutralize this apparent gain.

155. Performance in Heat Units.—The increasing use of superheated steam makes the use of heat units particularly desirable in stating engine performance. For this purpose, it is necessary to know the thermal condition of the steam when it enters and when it leaves the engine. The pressure, temperature and entrained moisture of the steam should be measured just above the throttle valve, and again as close as practicable to the exhaust flange. The difference between the heat units, entering and leaving, will show the amount converted into work. For example, the engine test represented by the curve in Fig. 161 gave the following values:

	½ load.	¾ load.	Full load.
Gage pressure	116.4	117.8	115.7
Quality of steam, per cent	98.4	98.4	98.4
Vacuum, inches	21.5	21.5	21.3
Total i.h.p.	94.4	129.3	173.5
Total wet steam, pounds per hour	1753.0	2296.0	3050.0

Assuming the exhaust steam to be dry, the following quantities are found:

	½ load.	¾ load.	Full load.
Heat above 32 deg. in B.t.u. per pound at entrance	1174.1	1174.3	1174.
Heat above 32 deg. in B.t.u. per pound at exit, 4.2 lb. press.	1129.2	1129.2	1129.1
Heat units converted into work, per pound	44.9	45.1	44.9
Total B.t.u per hour converted into work	78700.0	103550.0	136900.0
B.t.u. per i.h.p. per minute	13.90	13.31	13.16

If entering steam had been superheated, it would have been only necessary to add the heat due to this condition, in calculating the total heat. As another example, take the case of a cross-compound, single valve engine, 15½ and 26 by 15 running non-condensing.

The data of the tests at three different loads are as follows:

Revolutions	254.0	250.5	253.2
i.h.p.	150.8	198.7	256.6
Total steam per hour, pounds	3731.0	4702.5	5887.5
Pressure at engine, gage	124.4	125.2	124.5
Receiver pressure, gage	31.7	38.0	48.1
Back pressure in exhaust pipe	.13	.25	.28
Calorimeter data, deg. Fahr.	276.1	276.5	276.5
Manometer reading, pressure	.67	.76	.7
Moisture in steam, per cent	1.2	1.4	1.2
Dry steam per hour, pounds	3686.0	4636.7	5816.8
Electrical load, Amperes	380.7	519.3	674.5
" " Volts	250.0	250.0	250.0
" " e.h.p.	127.5	174.0	226.0
Dry steam per i.h.p. per hour, pounds	24.44	23.24	22.67

Applying the same method as in the preceding example:

	½ load.	¾ load.	Full load.
Heat above 32 deg. in B.t.u. per pound at entrance	1179.0	1177.4	1179.0
Heat above 32 deg. in B.t.u. per pound at exit	1146.7	1146.9	1146.9
Heat units converted into work	32.3	30.5	32.1
Total B.t.u. per hour converted into work	120500.0	143400.0	189000.0
B.t.u. per i.h.p. per minute	13.3	12.0	12.2
B.t.u. per e.h.p. per minute	15.8	13.7	13.9

156. Friction Losses.—In determining the size of engine suitable for a given work, it is important to know how much of the indicated horse power is available for useful work and how much is lost in the friction of the engine itself Many brake tests of engines have been made to determine this point, and the results are rather confusing. While the friction loss is rarely con-

stant, it certainly does not vary in proportion to the load; in fact, it is frequently less at full load than at light loads. In testing engines directly connected to electric generators, it is usually necessary to use the electric rather than the brake horse power, and the losses then include those in engine and generator. By running the engine with generator idle, a fair approximation to the friction of the engine may be obtained.

The following results are summarized from some tests which are quoted in French's Steam Turbines, pp. 180-81.

Type.	No. of Cylinders.	i.h.p.	Combined Efficiency.	Friction.	
				H.P.	Per Cent.
Vertical Compound	Three.	5400	94.5 95.2	118.6	2.2
Horizontal Compound	Three.	3500	89.5 92.9	266.3	7.6
Horizontal Cross-compound	Two.	2500	90.0		
Horizontal Cross-compound	Two.	1000 850 627 340	94.0 93.0 90.0 83.0	45.0	5.3
Compound-condensing	Two.	347 185		44.0 40.0	
Simple Non-condensing	One.	142 84			12.7 8.4

Engines may then be expected to have an efficiency at full load of from 90 to 95 per cent and perhaps from 5 to 10 per cent less when combined with an electric generator. Vertical engines develop considerably less friction than horizontal, on account of the small amount of piston friction.

157. Summary of Performance.—Mr. French in his book on Steam Turbines, before alluded to, has tabulated the results of tests on various types and sizes of engines. Table XIX., p. 194, gives the best results obtainable, while Table XX., p. 195, is one of the average performances. The results in this second table were originally published in Barrus' Engine Tests and represent the

performance of 23 compound condensing engines, under ordinary commercial conditions.

Mr. French also shows that with engines of this type a variation of from 55 to 60 per cent of the maximum power can be allowed without increasing the water rate more than ten per cent.

CHAPTER XIV.

STEAM ENGINE DESIGN.

158. In General.—The principles governing the design of certain special mechanisms peculiar to steam engines, such as valves, links, governors, etc., have been explained in previous chapters. The details of rods, journals, bolts, slides, etc., may be worked out by the same rules as govern the design of similar parts in other machines. Like most special machines, however, the steam engine has developed many features which cannot be treated in this way; they are the result of a process of selection due to long experience on the part of builders; the size and the shape of these details cannot be calculated by the ordinary rules for strength and stiffness, but must be determined by a study of existing designs. Examples of many such details will be given in the present chapter, together with both empirical and rational rules for their design.

The empirical formulas given in this chapter are taken mainly from a paper published by Prof. John H. Barr, formerly of Cornell University.* They were obtained by comparing the actual dimensions in use on modern engines, plotting curves to connect these dimensions with the horse power, speed, etc., of the engines, and then deriving a formula from an average curve. In all, 165 engines were represented, made by 25 different firms. Of these, 80 were classed as high-speed and 85 as low-speed engines, reference being had to rotative speed. The engines were all simple and the maximum pressure was assumed at 100 lb. gage.

The engines ranged in size from 20 to 240 horse power for the high-speed and 45 to 740 horse power for the low-speed.

The notation used is as follows:

D = diameter of piston.
A = area of piston.
L = length of stroke.

* Transactions, A. S. M. E., Vol. XVIII.

V = piston speed in feet per minute.
S = steam pressure.
N = revolutions per minute.
C or K = any constant.
Dimensions in inches, unless otherwise stated.

NOTE.—The low-speed engines were side-crank, having one main journal. The high-speed were center-crank with two main journals.

There is at the present time a tendency to increase the principal dimensions of steam engine details to allow for the higher initial pressures generally employed. Some builders are getting the same result by using smaller cylinders for the higher pressures, keeping the rating of the engine the same as before. When pressures higher than 100 lb. are employed, the formulas in the following articles should be revised in accordance; that is, the constant used should be increased to correspond with the new pressure. Attention will be called to some of these changes in succeeding articles.

159. The Frame.—Engine frames may be classified first as vertical, or horizontal, the latter being much the more common. Vertical frames are usually of the A type with the cylinder at the top and the shaft at right angles to the plane of the A. Fig. 168 illustrates a vertical cross-compound engine of the Corliss type, having a frame of this character. The straight lines used and the symmetry of the arrangement with reference to the lines of force relieve this frame from bending and torsion and only tensile stresses need to be calculated.

In small engines of the marine type, the cast-iron is sometimes omitted in the front leg of the A, and replaced by two steel rods running, in an inclined position, from the cylinder to the bed plate. This construction gives easier access to the moving parts, but makes it necessary to use a single guide for the crosshead.

The A frame is occasionally used in horizontal engines, as for example, in the well-known Straight Line engine, but in this case the legs of the A run from the cylinder to the main bearings, and the shaft is in the plane of the A. Mechanically, this is a correct design, because the force of the steam is transmitted directly to

Fig. 168. Vertical Cross-Compound Engine.

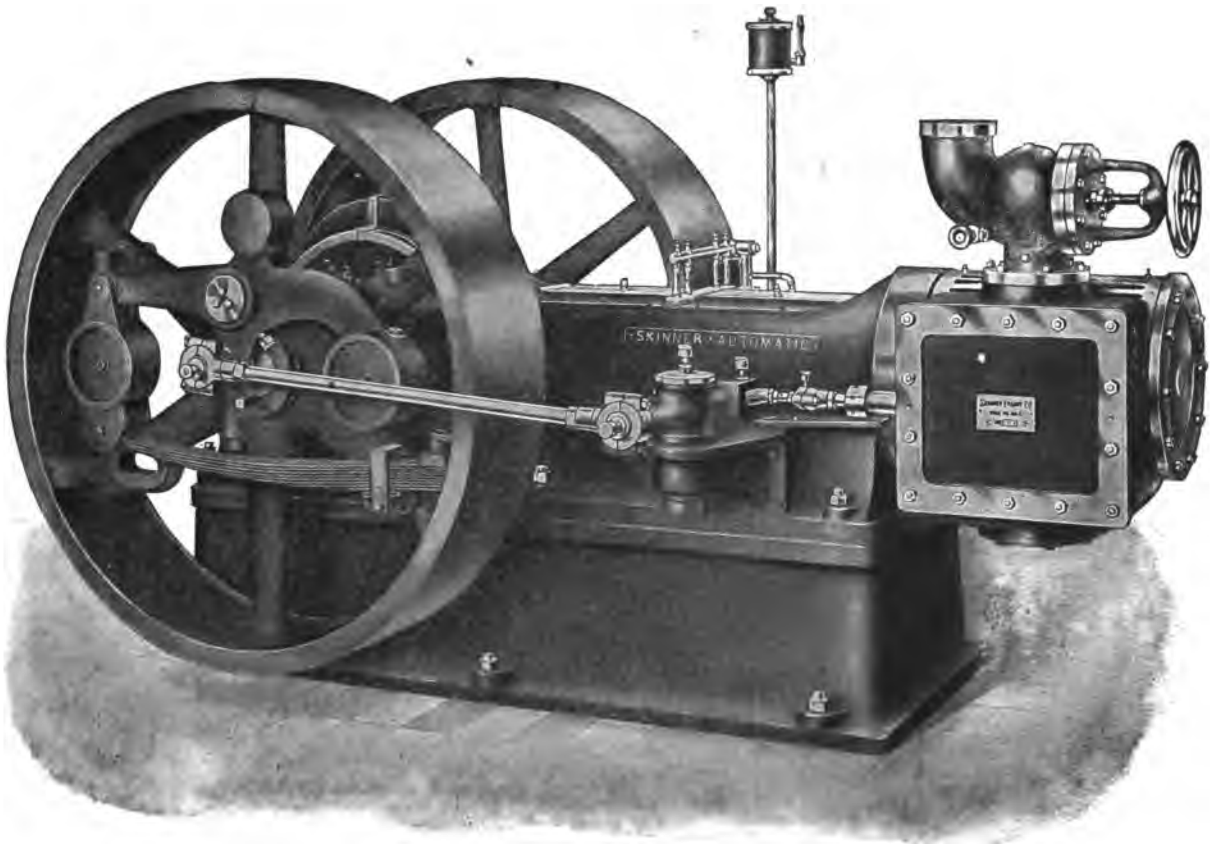


Fig. 169. Center-Crank High-Speed Engine.

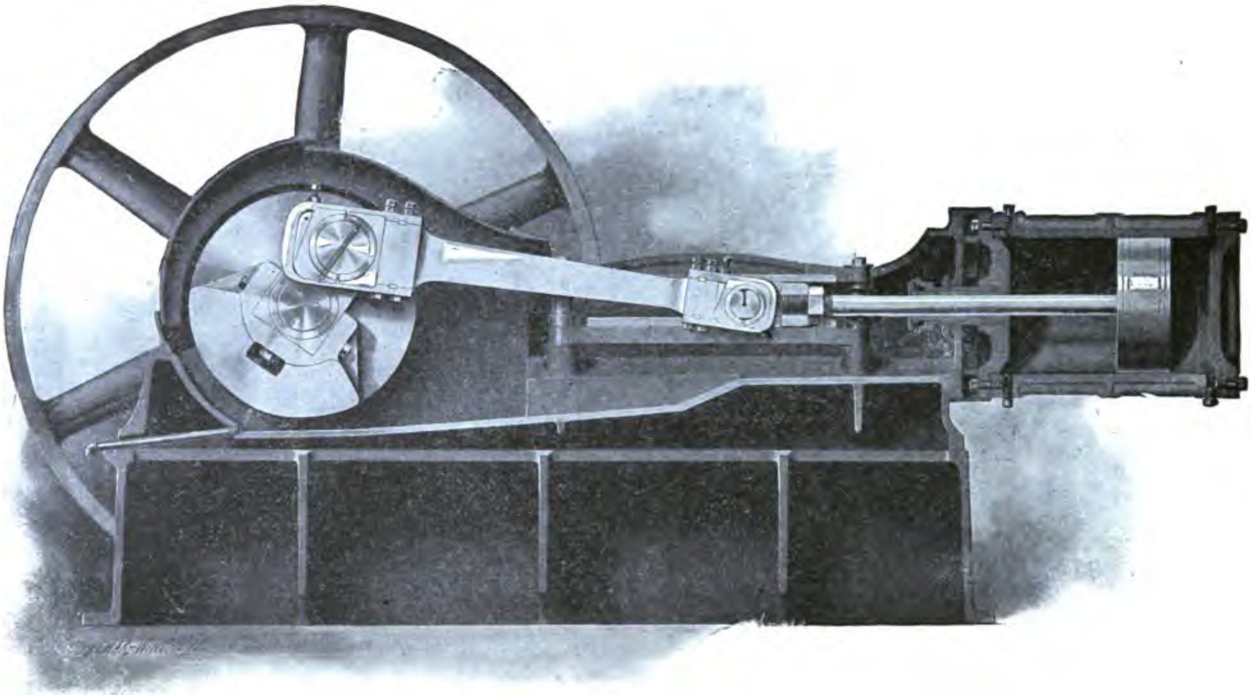


Fig. 170. Section Showing Construction of Frame and Base of Side-Crank Engine.

the bearings along the straight sides of the frame, and there are no bending or twisting moments in the structure.

Horizontal engines may be either center-crank or side-crank (see Art. 17) and the frame must be designed to correspond. Center-crank engines are usually of small or medium size with overhanging cylinders and horizontal crossheads. (See Fig. 169.) Such a frame is relatively massive and has an excess of strength. Care should be used, however, in designing the neck and flange at the cylinder end, to secure sufficient strength to allow for the overhang of the cylinder.

Fig. 170 illustrates the design of frame for a small side-crank engine with cast-iron sub-base and overhanging cylinder. The sub-base is preferable to brick or concrete for engines of this size,

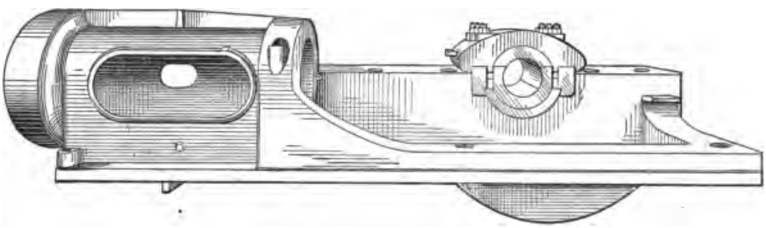


Fig. 171. Frame for Large Side-Crank Engine.

as it is more durable and easier to keep clean. Masonry is better kept down to the floor level. Fig. 171 shows a frame for a side-crank engine of large size, with a barrel body and guides for a vertical crosshead. Notice the reinforcement of the frame under the main bearing and in front of the crank. Owing to the fact that the main journal is at one side of the center line of the engine, the frame is subjected to bending and torsion near the crank, and especial care should be taken to get plenty of metal here.

The principal stress at the guides is caused by the vertical thrust of the connecting rod, and the tubular form is well adapted to stiffen the frame at this point. In an engine of this size, the cylinder rests on supports of its own. The light, open girder

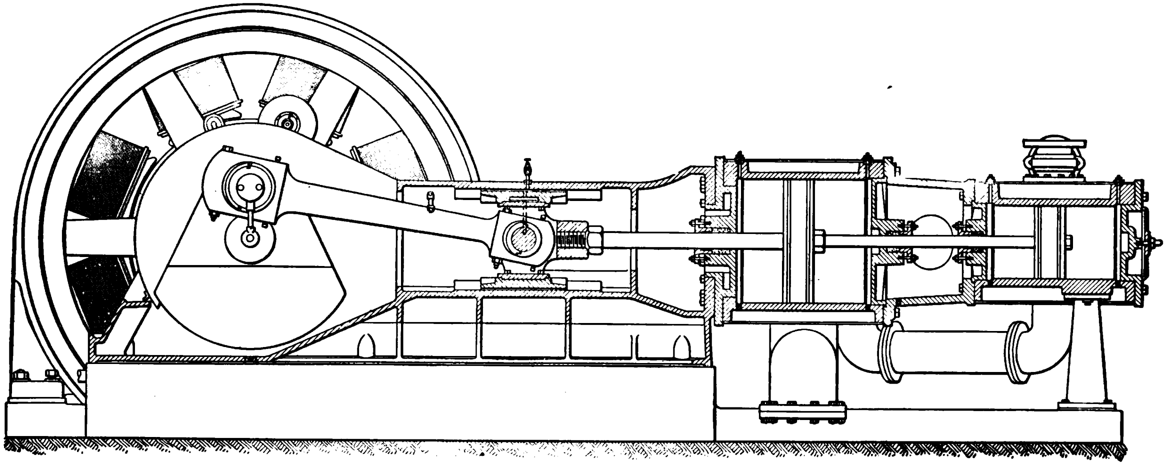


Fig. 172. Longitudinal Section, Tandem-Compound Engine.

frames, formerly so much used with the Corliss type of engine, are being replaced by these heavier forms.

The cross-compound engine calls for no new features in frame design, being merely a right and left engine coupled together. The tandem compound has a short frame or yoke between the two cylinders, as shown in Fig. 172, of sufficient length to accommodate the two stuffing boxes and to afford easy access to them. The low-pressure cylinder is supported by its exhaust pipe, and the high-pressure on a special stand.

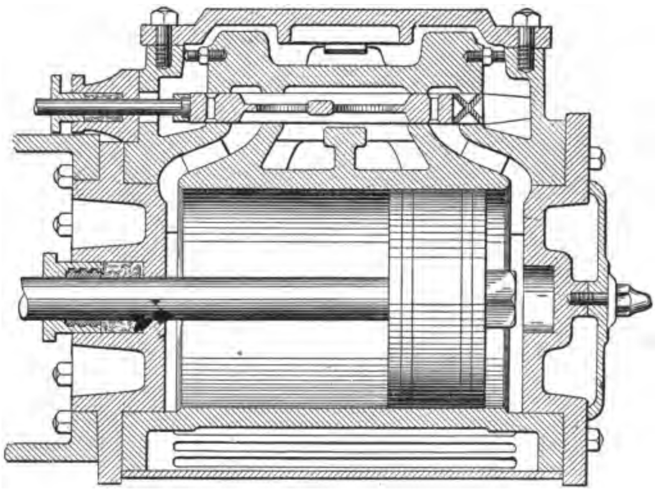


Fig. 173. Section of Cylinder, Slide-Valve Engine.

160. The Cylinder.—As has been noticed in the preceding article, the cylinders of small and medium engines usually overhang, while those of larger engines have independent supports. In either case, the crank end of the cylinder is bolted directly to a flange on the end of the frame. (See Figs. 170 and 172.)

Fig. 173 illustrates the ordinary proportions and construction of the cylinder for a slide valve engine, while Fig. 174 shows the more complicated design when Corliss valves are used. It is impossible in one short chapter to notice the variety of details entering into the design of engine cylinders and steam chests, and only the most important ones will be here considered.

The ratio of diameter of bore to stroke of piston varies from $\frac{1}{1}$ to $\frac{1}{3}$, being greatest in high-speed and least in low-speed engines. The clearance volume is about 5 per cent of the piston displacement in long-stroke engines and sometimes as much as 10 per cent in high-speed engines having slide valves and short strokes.

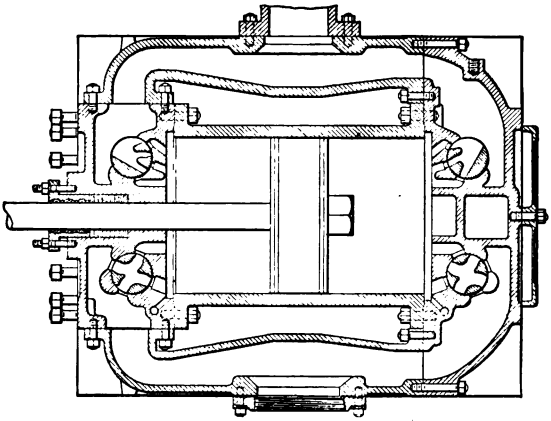


Fig. 174. Section of Cylinder, Corliss Engine.

The piston speed of high-speed engines varies from 530 to 660 ft. per minute with an average of 600, while in low-speed engines 500 and 850 ft. per minute are the limits, the average still being 600 (Barr).

The counterbore at the ends extends far enough in so that the piston rings slightly overlap the edge at each end of the stroke, thus preventing the formation of shoulders. The heads are turned to fit the counterbore, and should be provided with set-screws at the flanges to force them off.

161. Thickness of Cylinder Walls.—A discussion of the thickness of steam cylinders will be found in the author's book on Machine Design, p. 46, where the following formula is derived:

$$t = \frac{pd}{8000} + \frac{d}{200}\sqrt{p + \frac{p^2}{1600}}$$

where $d =$ internal diameter,

$p =$ maximum pressure.

Professor Barr gives the following empirical formula for the cylinders of low-speed engines: $t = CD + 0.3$, where C ranges from 0.04 to 0.06, with an average value of 0.05. (Dimensions are in inches.)

The thickness of flanges usually exceeds this by about 20 per cent.

The number of bolts for fastening the head is usually about $0.7D$, and the diameter is given by the formula

$$d = \frac{D}{40} + \frac{1}{2} \qquad \text{(Barr)}$$

NOTE.—The factor of safety of 10 or 12 is generally used in engine design.

162. The Piston.—Pistons up to 12 inches in diameter are usually made of cast iron, with two cast-iron rings sprung into place. The pistons are hollow and are turned true all over. The ends are preferably flat, to match the heads of the cylinder and reduce clearance.

The rings are from ⅜ to ⅝ in. wide, are bored eccentrically and are split on the thin side. If we call the cylinder bore D, the dimensions of the ring may be as follows:

$$\left.\begin{array}{ll}\text{Outer diameter} & = 1.05D \\ \text{Inner diameter} & = 0.97D\end{array}\right\} \text{Before springing together.}$$

$$\begin{array}{ll}\text{Greatest thickness} & = 0.05D \\ \text{Eccentricity} & = 0.01D\end{array}$$

The thin part of the ring will need to be tapered slightly on the inside for about 30 deg. on either side of the split. These proportions will give approximately a uniform pressure on the entire circumference. Some builders prefer to fill in about 60 deg. of the groove on the lower side of the piston and only use rings of 300 deg. The weight of the piston will keep it tight at the bottom. The ratio of length of piston to diameter is as follows:

High-speed Engines:

Face $= 0.30$ to $0.60D$; average $= 0.46D$.

Low-speed Engines:

Face $= 0.25$ to $0.45D$; average $= 0.32D$. (Barr.)

On large engines the piston is usually a steel casting and is made as light as possible with a face narrow in proportion to the diameter, and sectional packing held in place by springs. (See Fig. 175.) In such case, the cylinder heads are cast to match the piston, thus reducing clearance. Pistons are generally secured to the rods by straight or taper fits and by check nuts. (See Fig. 175.)

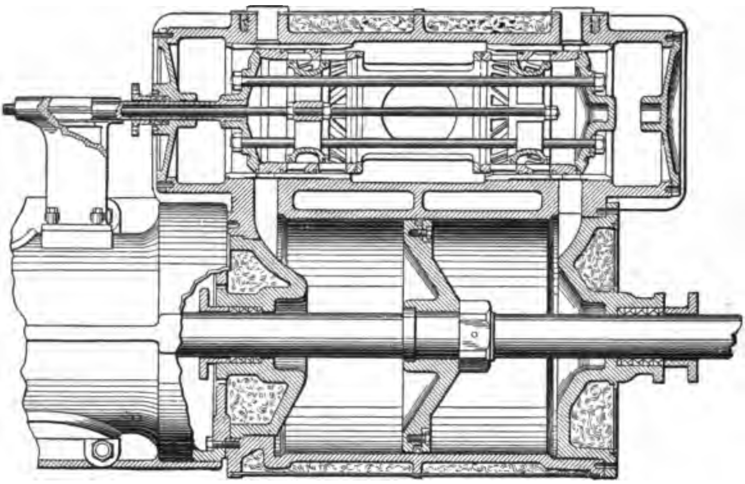


Fig. 175. Section of Large Cylinder Showing Packing and Shape of Piston and Heads.

The rod is of steel and may be calculated by the usual formulas for columns, regarding one end as fixed and one as round. Empirical formulas for the diameter are as follows:

High-speed Engines:

$d = 0.12$ to $0.175\sqrt{DL}$; average $= 0.145\sqrt{DL}$. (Barr.)

Low-speed Engines:

$d = 0.10$ to $0.13\sqrt{DL}$; average $= 0.11\sqrt{DL}$ when D and L are the diameter and stroke of cylinder in inches. If the rod is threaded at either end, the area at root of thread must be sufficient to withstand the direct tension.

The length of rod between shoulders is from 1.1 to 1.2L.

If pressures over 100 lb. per sq. in. are to be used, the formula for diameter of piston rod should read:

$d = 0.17\sqrt{DL}$ for high-speed engines.

$d = 0.14\sqrt{DL}$ for low-speed engines.

as more closely representing modern practice.

The stuffing box may contain soft packing of asbestos and rubber or metallic packing made of some alloy. The latter is more suitable for steam over 400 deg. Fahr.

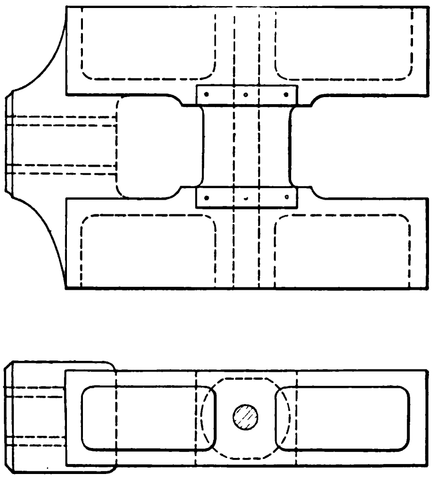


Fig. 176. Crosshead, Porter Allen Engine.

163. The Crosshead.—Crossheads may be classified as horizontal and vertical; the horizontal crosshead has guides in the same horizontal plane either side of the pin; the vertical crosshead has guides above and below the pin.

The slipper crosshead is of the vertical type with the upper guide absent. Fig. 170 shows a horizontal crosshead in position and Fig. 172 a vertical one. The type and shape are largely determined by the class of frame adopted for the engine.

Fig. 176 shows a horizontal crosshead, as used on the Porter Allen engine, consisting of a hollow steel casting, with the pin in-

serted in shallow, vertical grooves and secured by a central bolt running through pin and head. If the pin is cast with the head, the hub which holds the piston rod is made removable so as to allow of finishing the pin in a lathe. The guides are straight, narrow metal bars, as illustrated in Fig. 170.

Fig. 177 shows one style of vertical crosshead such as is used with frames having a circular guide barrel. The hub and head are cast in one piece and the pin is inserted at the side, being pro-

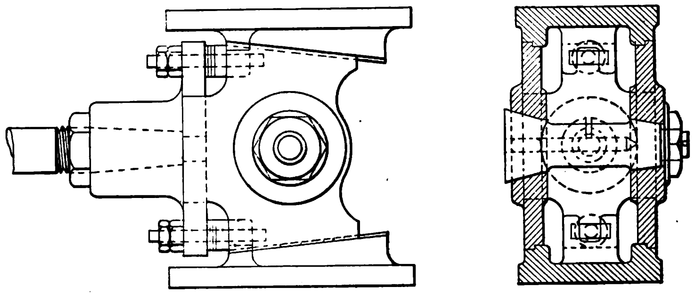


Fig. 177. Vertical Crosshead.

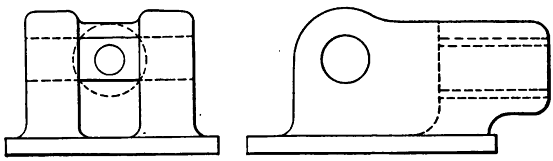


Fig. 178. Slipper Crosshead.

vided with two taper fits and a lock nut. The pin may be flattened at top and bottom, or may be so arranged that it can be rotated 90 deg. when it has worn on two sides. The slides are fitted on inclined ways, and can be adjusted for wear by means of screws and nuts as shown. The cross section of the bearing surfaces of the slides may be rectangular, circular or of flat V shape.

Fig. 178 represents a one slide or "slipper" crosshead. This form is not much used in this country, as it is not symmetrical and does not guide the rod as well as those having double slides.

A crosshead is secured to the piston rod by a screw and check nut, or by having the hub split and clamped to the rod by transverse bolts. It is well to have some means of adjusting the rod endwise in the head, so as to equalize the clearance of the piston at the two ends of the cylinder.

The pin (wrist pin it is generally called) should be located centrally with respect to the slides. The vertical pressure due to the obliquity of the connecting rod is exerted at the pin, and if this should be near one end of the slides, the latter would wear unevenly. The slides are frequently faced with Babbitt metal.

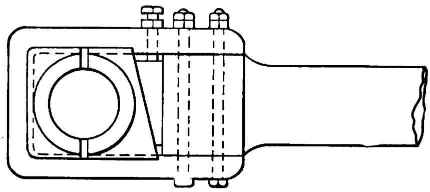


Fig. 179. Connecting Rod End with Strap.

According to Barr's investigations, the following are the proportions used in common practice:

High-speed Engines:

Area of slide = 0.45 to 1.6A; average = 0.63A.

Average pressure on slide = 10.5 to 38 lb.; average = 27 lb. per square inch.

Projected area of wrist pin = 0.06 to 0.11A; average = 0.08A.

Proportions of wrist pin, l = 1 to 2d; average = 1.25d. (A = area of piston in square inches.)

Low-speed Engines:

Area of slide = 0.32 to 0.62A; average = 0.46A.

Average pressure on slide = 29 to 58 lb.; average = 40 lb. per square inch.

Projected area of wrist pin = 0.054 to 0.10A; average = 0.07A.

Proportions of wrist pin, l = 1 to 1.5d; average = 1.3d.

The guides should be of such lengths that the slide may overlap considerably at each end of the stroke. It will do no harm if the center of the wrist pin comes nearly to the end of the guide.

164. The Connecting Rod.—The connecting rod may be a steel casting, a wrought-iron forging, or it may be planed out of bar steel. The cross section is usually rectangular, but an oval or an I section may be used when the rod is cast. The wrist-pin end of the rod is generally so arranged with a strap and brasses that it may be disconnected without removing the pin from the crosshead. The crank-end of the rod is usually solid with inserted brasses, except on center crank engines where this is impracticable. Fig. 179 shows an end with a strap fastened by bolts,

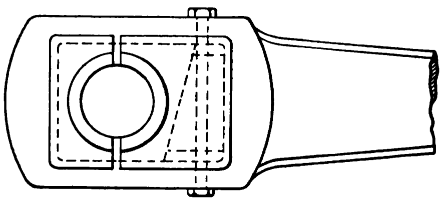


Fig. 180. Solid End.

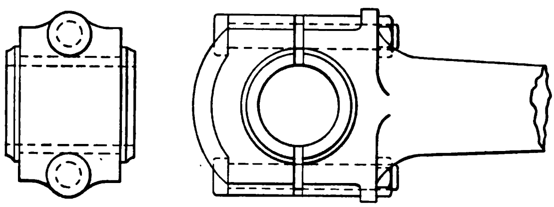


Fig. 181. Marine Type End.

so that it can be readily disconnected. The brasses are tightened by a wedge operated by a bolt and check nut. The comparatively thin gib and cotter are not so much used as formerly, as they do not give so good a bearing on the brass.

Fig. 180 illustrates a rod with a solid end suitable for a side crank engine. No bolts are necessary, and the brasses are tightened by a wedge, as in the preceding figure. The solid end is better, as it is more compact and has fewer pieces to work loose. Large center crank engines frequently have rods with ends of the marine type shown in Fig. 181. This rod can be disconnected

and the brasses removed by taking off the cap as in an ordinary journal bearing.

Connecting rods were formerly made with a circular cross section and largest midway of the length. This shape is now seldom used, and the rods are largest near the crank end, tapering uniformly towards the crosshead. The dimensions of the section at the middle may be calculated by the column formulas; in the plane of vibration, the rod may be regarded as a column with round ends, but at right angles to this plane, the flexure will be that of a column with both ends fixed. This indicates a depth twice the breadth at this point.

Professor Barr gives the following:

High-speed Engines:

Length of rod $= L_2 = 3L$.

Breadth of rod at center $= 0.045$ to $0.07\sqrt{DL_2}$; average, $0.057\sqrt{DL_2}$.

Depth of rod at center $= 2.2$ to $4b$; average $= 2.7b$. (L and D as before.)

Low-speed Engines:

Length of rod $= L_2 = 2.75L$.

Diameter of round rod at center $= 0.082$ to $0.105\sqrt{DL_2}$; average $= 0.092\sqrt{DL_2}$.

The strap and bolts, if any, may be calculated by the usual rules for tension and shear.

In designing a complete rod, it is best to so arrange the ends that the wear will tend to lengthen the rod at one end and shorten it at the other. (See Fig. 170.)

165. The Crank.—Side cranks are usually made of a separate casting or forging, keyed to the end of the shaft and having an inserted crank pin. The same rules are applicable as in the case of any crank or lever. The following formulas are taken from the author's book on Machine Design.

Let $l =$ radius of crank.
$t =$ thickness of web at center.
$h =$ breadth of web at center.
$d =$ diameter of eye.

d_1 = diameter of pin.
b = breadth of eye.
D = diameter of hub.
D_1 = diameter of shaft.
B = breadth of hub.
P = pressure on pin.

Then,

$$th^2 = \frac{3\,Pl}{S}$$

$$B\,(D^2 - D_1^2) = \frac{6\,Pl}{S}$$

$$b^2\,(d - d_1) = \frac{6\,Px}{S}$$

where x is the distance from center of pin to center of eye measured along the axis. From these formulas the necessary dimensions may be calculated.

Fig. 182 shows such a crank, suitable for a low-speed engine. Engines having a high rotative speed need a counterbalance on the crank. (See Chapter IX.)

This is sometimes cast as a part of the crank, as in Fig. 172 and sometimes attached by bolts or keys, as in Fig. 170. Center cranks are best forged as an integral part of the shaft, as shown in Fig. 183 and the pin made of the same diameter as the journals. The maximum bending moment in such case is at the center of the pin, the whole shaft being in the condition of a simple beam with a load at the center, and a length equal to the distance from center to center of journal. If a counterbalance is necessary, it can be made of cast iron of some such shape, as shown in Fig. 184, and one of these secured to each crank by keys or bolts. The crank disc is thus relieved from any stress save that due to centrifugal force.

In designing any form of side crank, it is well to keep the overhang as short as possible, and thus reduce the bending moments on the shaft and on the frame near the shaft. The most serious objection to an overhung crank is the fact that the thrust on the pin is not in line with the resistance of the journal bearing. Fig. 185 shows the extent to which this distance may be reduced in a steel disc crank by a proper proportioning of the parts.

The dimensions of the crank pin are determined by several considerations:

(a) It should be of sufficient length to prevent heating.

(b) It should have sufficient projected area to prevent the pressure from squeezing out the lubricant.

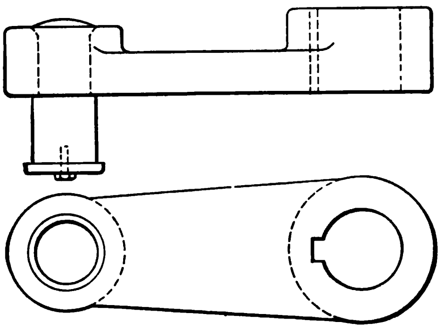


Fig. 182. Overhung Crank.

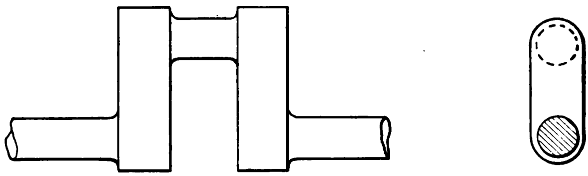


Fig. 183. Center Crank.

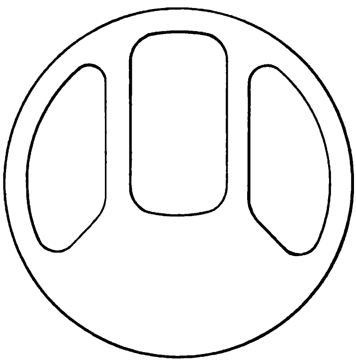


Fig. 184. Counterbalance for Crank.

(c) It should have a diameter such as to insure strength and stiffness.

It has been generally assumed by engine designers that a long pin is necessary to prevent heating; this assumption is based on rational grounds,* but there is reason to doubt its truth, and some recent experiments have seemed to discredit it.

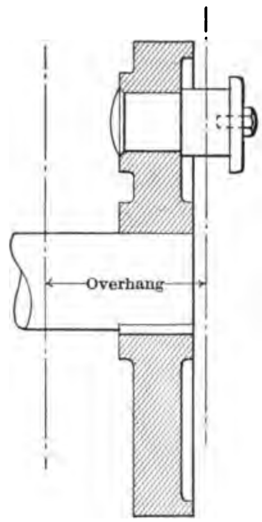


Fig. 185. Overhung Disk Crank.

The projected area should be enough to limit the average pressure so that it shall not exceed 500 lb. per square inch. The strength of the pin is calculated as that of any cantilever or simple beam, regarding the load as concentrated at the middle of its length. In calculations for wear or heating, the average pressure should be used, but in all calculations for strength, the maximum pressure.

Barr's formulas for crank pins are as follows:

High-speed Engines:

Length $= 0.13$ to $0.46 \dfrac{HP}{L} + 2.5$; average $= 0.3 \dfrac{HP}{L} + 2.5$.

* Benjamin's Machine Design, p. 104.

Projected area $= 0.17$ to $0.44A$; average $= 0.24A$.

Low-speed Engines:

Length $= 0.4$ to $0.8\frac{HP}{L}+2$; average $= 0.6\frac{HP}{L}+2$.

Projected area $= 0.065$ to $0.115A$; average $= 0.09A$.

It will be seen from the variation in the coefficients that engineering practice in the matter of crank-pin dimensions has not yet crystallized.

For initial pressures greater than 100 lb. per sq. in., the following formulas represent modern practice in the matter of crank pins more closely than those given above:

$$\text{Length} = 0.3\frac{HP}{L} + 2 \text{ in. for high-speed engines.}$$

$$\text{Length} = 0.5\frac{HP}{L} + 2 \text{ in. for low-speed engines.}$$

Projected area $= 0.12\ A$ for high-speed engines.
Projected area $= 0.11\ A$ for low-speed engines.

These values are for side-crank engines. The exceptionally large values for projected area given in Barr's formula are due to the fact that most of the high-speed engines figured on have center cranks. It will be understood that the crank pin of the center crank must be as large as the shaft itself since the principal bending moment comes at the center of the pin.

166. The Shaft and its Bearings.—The main journal of the shaft of a side-crank engine is subjected to a twisting moment $T =$ the pressure on crank pin $\times$ radius of crank, and to a bending moment $M =$ the same pressure $\times$ the overhang. (Fig. 185.)

These should be combined by the formula

$$T^1 = M + \sqrt{M^2+T^2}$$

and the result T^1 be used as an equivalent twisting moment. The diameter of journal may then be calculated by the usual formulas for shafts. If a heavy fly wheel is used, the bending moment due to its weight may be combined with the turning moment, and the diameter of the shaft under the wheel be determined. In slow-

speed engines, this diameter is frequently greater than that of the journal.

Center-crank engines usually have a maximum bending moment at the middle of pin, and there is no twisting moment at this point. If there are overhung belt wheels or fly wheels, there will be a combined bending and twisting moment at the center of each journal.

The limit of pressure per square inch of projected area of an engine journal is determined by combining the pressure of the steam and the weight of the shaft and wheel. It is usually lower than for crank pins.

Empirical formulas for the proportions of the main journal are as follows (Barr):

High-speed Engines:

Diameter of journal $= 6.5$ to $8.5\sqrt[3]{\frac{HP}{N}}$; average $= 7.3\sqrt[3]{\frac{HP}{N}}$.

Length of journal $= 2$ to $3d$; average $= 2.2d$.

Projected area of journal $= 0.37$ to $0.70A$; average $= 0.46A$.

Low-speed Engines:

Diameter of journal $= 6.0$ to $8.0\sqrt[3]{\frac{HP}{N}}$; average $= 6.8\sqrt[3]{\frac{HP}{N}}$.

Length of journal $= 1.7$ to $2.1d$; average $= 1.9d$.

Projected area of journal $= 0.46$ to $0.64A$; average $= 0.56A$.

If initial pressures of more than 100 lb. per sq. in. are used, the following formulas give more nearly the present practice in proportioning main journals:

For high-speed engines:

$$\text{Diameter of journal} = 8.5\sqrt[3]{\frac{HP}{N}}$$

For low-speed engines:

$$\text{Diameter of journal} = 7.5\sqrt[3]{\frac{HP}{N}}$$

The most important function of an engine bearing is that of adjustment for wear. In a vertical engine, this is a simple matter, as the wear is always in one direction, but in a horizontal engine, wear is caused by the horizontal pressure of the steam and by the vertical pressure due to the weight of shaft and wheel. To meet

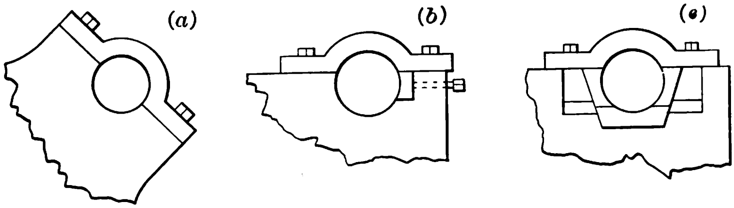


Fig. 186. Types of Main Bearings.

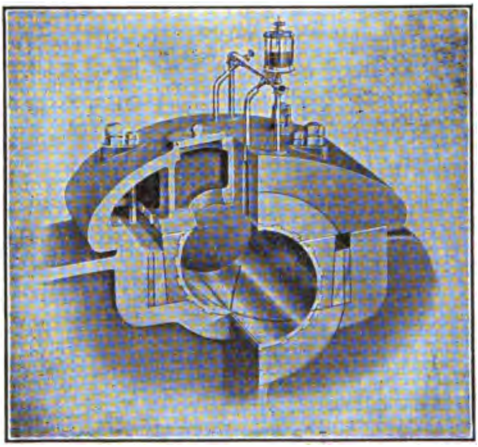


Fig. 187. Details of a Four-Part Bearing.

this, there should be adjustment in more than one direction. Two-part, three-part and four-part bearings have all been used for this purpose. (See *a*, *b* and *c*, Fig. 186.) The split of the two-part bearing is inclined so that the adjustment may compensate in part for both kinds of wear. The other two are more

correct, as far as adjustment is concerned, but sometimes give trouble from heating, if the cheek pieces are set up too tight. Figs. 187 and 188 show details of modern four-part bearings. The outboard bearing of a side-crank engine is a plain pedestal bearing with vertical adjustment.

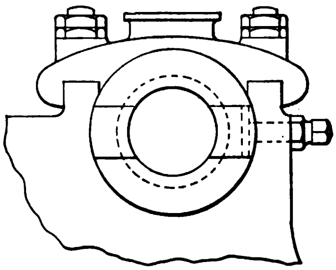


Fig. 188. Another Design of Four-Part Bearing.

167. The Fly Wheel.—The proper weight of rim for a fly wheel can be determined by the methods indicated in Chapter IX. Such wheels up to 8 or 10 ft. in diameter may be cast in one piece, but are frequently in halves to facilitate putting them on the shaft after the engine is in place. The two halves may

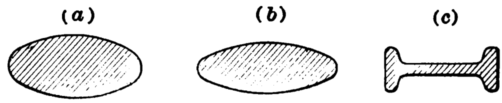


Fig. 189. Sections of Wheel Arms.

even then be cast together with comparatively thin webs of metal connecting them at rim and hub. The wheel can be bored and turned as one piece and then broken apart with wedges. This leaves a tight, reliable joint not readily displaced. Wheels from 10 to 16 ft. in diameter should be cast in two pieces for convenience in transportation. Wheel rims of still larger diameter should be made in segments, one for each arm, the arms being bolted to rim and hub.

The number of arms varies from six in the smaller wheels to twelve or more in those from 20 to 30 ft. in diameter. The cross section of the arm is generally elliptical or lens-shaped (a) and (b) Fig. 189. The I-section is more economical of material, but offers considerable resistance to the air when in rapid motion.

The principal stresses in the arm are compression, due to the weight of the rim, and tension, due to the same cause and centrifugal force. If a fly wheel is also used for a belt wheel, the arms are subjected to bending. Investigations made by the author* have shown that the bending moment, at the hub, is approximately double that at the rim and that the maximum bending moment on the arm which happens to be nearest the pulling side of the belt may be determined by dividing the total turning moment by one half the number of arms.

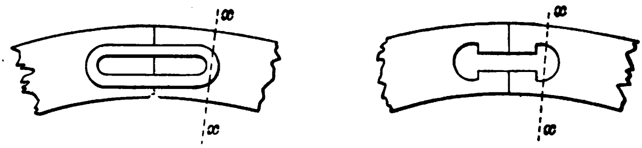


Fig. 190. Fly-Wheel Rim Joints.

In wheels having thick rims, this bending moment will be less. The arm should be designed for the maximum moment at the hub, and then tapered so that the section modulus at the rim shall not be less than half that at the hub.

Arms of very large wheels can be made hollow. The weakest points in a large wheel are the joints in the rim. Rims which have considerable depth in proportion to their breadth, such as those used on the wheels of blowing engines, rolling mill engines and direct-connected electrical units, offer little difficulty in this respect. The joints are usually fastened by links or T-headed "prisoners," as shown in Fig. 190.

* American Machinist, Sept. 22, 1898.

The links are of steel, are put in place while hot, and by shrinking, pull the two halves of the joint tightly together Such a joint is not exposed to much bending, and the stress is direct tension. The weakest section is at xx and the joint will fail there or by shearing of the lug inside the link.

In case a rim is hollow or is channeled as in (c) Fig. 189, the strength of this form of joint may be made nearly 100 per cent, by properly proportioning link and rim section.

Thin rims, such as are used on wheels for belt transmission, are much more difficult to join. The centrifugal force of the rim between the arms puts it in the condition of a beam fixed at the

Fig. 191. Broken Rim; Joint of Ordinary Type.

ends and uniformly loaded and causes a maximum bending moment midway between the arms. The joint formerly used in such cases was particularly weak against such bending action, as may be seen by inspection of Fig. 191. This figure shows the appearance of the ordinary flanged joint after the wheel had been burst by centrifugal force. The inner edge of the flange acts as a fulcrum, and the joint is forced open by the outward pressure. It is essential for the safety of a joint in a thin rim that it should be located over an arm and that the tensile strength of the joint should be as near that of the whole rim as possible.

Fig. 192 illustrates the construction of a joint in the rim of a combined belt and fly wheel for an Allis-Corliss engine. The flange bolts are brought close to the rim, and the bolts, which fasten the arm to the rim, help to strengthen the latter.

Fig. 193 shows a rim of channel section with a reinforced joint and T-headed links, one to each channel. This construction brings the links near the center of gravity of the cross section, and if properly proportioned, the joint will be practically as strong as the rest of the rim.

The weight of fly-wheel rims in high-speed engines varies from

$$650{,}000{,}000{,}000 \text{ to } 2{,}000{,}000{,}000{,}000 \frac{HP}{D_1^2 N^3}$$

with an average value of $1{,}200{,}000{,}000{,}000 \dfrac{HP}{D_1^2 N^3}$

where D_1 = diameter of wheel in inches,
N = number of revolutions per minute.

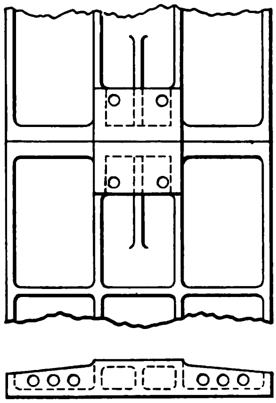


Fig. 192. Joint for Band-Wheel Rim Located at end of Arm.

168. Lubrication.—The design of valve motions, linkages and governors has been discussed in other chapters. One feature of engine design has become of particular importance since the introduction of electric transmission and continuous running, and that is positive lubrication.

The lubrication of the cylinder has been considered in Chapter XII. For the proper oiling of the moving parts outside of the cylinder, either gravity systems or a force pump may be employed. Perhaps, the most usual method is to supply the oil from a tank on top of the engine (Fig. 194), whence it flows by gravity to the various bearings. On leaving the bearings, it flows to a receiver in the bed of the engine and is pumped back to the tank by a force pump. The tank acts as a filter and removes dirt and

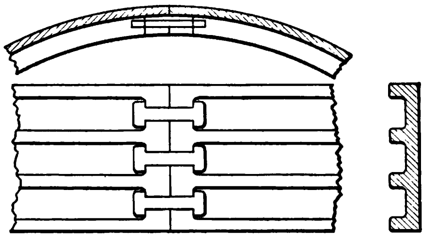


Fig. 193. Reinforced Joint Held by Links.

abraded metal from the oil, so that it may be used over and over.

To prevent splashing and waste of the oil, the reciprocating parts are enclosed as shown in the figure, and all oil is thus caught and returned to the tank. Where several engines are used, one tank and filter located above the engines will serve them all. Large engines frequently have independent pumps and oil tanks for the shaft journals and keep them flooded with a steady stream of oil.

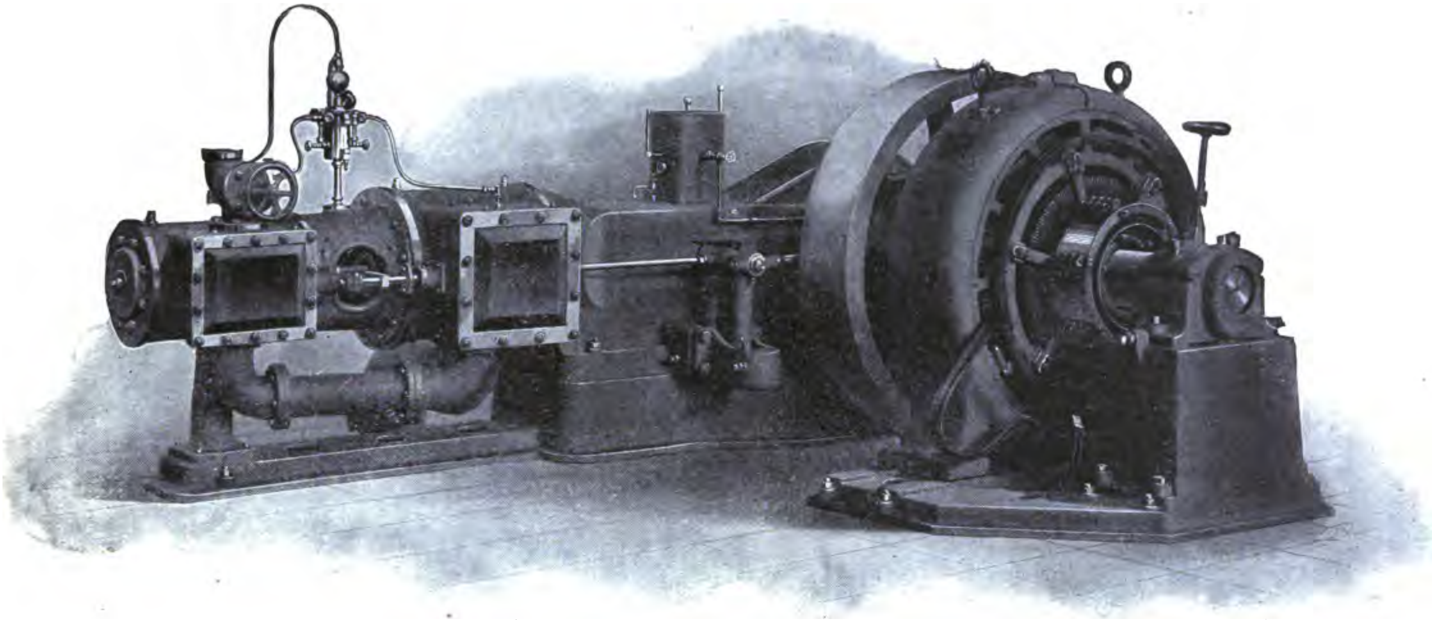


Fig. 194. View of Engine, Showing Oiling System.

CHAPTER XV.

SPECIFICATIONS AND COSTS.

169. Specifications.—The specifications for an engine may be prepared by the purchaser, and in the case of large engines, such as pumping engines and those used in central power plants, this plan is frequently followed, and the engines are built to order in accordance with the specifications. The more common plan is for the intending purchaser to indicate in a brief statement the principal requirements which are to be insisted upon, and to ask for bids on this basis. The detailed specifications are then furnished by the engine builder to accompany his bid, and when the bid is accepted, these specifications form a part of the contract.

The specifications, after giving a detailed description of the engine, should state particularly just what is included in the tender in the way of accessories and what share of the expense of erection will be borne by each of the parties to the sale.

The bid should state where the engine is to be delivered, either on the cars at the factory, at the plant of the purchaser or on the foundations ready for the connections. It is customary for the engine builders to supply a superintendent of erection for a limited time, the purchaser paying the expenses of the superintendent and furnishing the common labor. A guarantee may be given, if desired, covering the regulation and the steam consumption of the engine.

As an illustration of the form of specifications prepared by the intending purchaser, the following specifications for a pumping engine for a municipal plant are given in full.

Specifications for a 20,000,000 Gallon Vertical Triple Expansion Pumping Engine for Water Works.

Plans, Specifications and Sealed Proposals will be received at the office of the Board of Control, City Hall, until 12 o'clock m., ..., for furnishing and erecting in the engine house to be built at the pumping station, one vertical triple expansion, crank and fly wheel pumping engine, with a capacity of

twenty million United States standard gallons of water in twenty-four hours.

The pump to have a stroke of not less than 60 inches, with a piston speed of not over 225 feet per minute; and with a steam pressure at the boilers not exceeding 150 pounds per square inch. The engines to pump against a pressure of not more than 90 pounds per square inch on the pumps.

The pumping engine will be required to be in complete running order nine months from the day the bidder received written notice from the director of public works to begin work.

The pump end and connection to be made of sufficient strength to withstand a water pressure of 200 pounds per square inch.

The suction and discharge valves of the pump to be of a capacity to give a velocity not exceeding four feet per second.

The type of pump to be three single acting plungers.

The discharge valve deck to set not higher than two feet above city base of levels. City base of levels is about 3½ feet above ordinary lake level. All valve decks to be cast separate from the pump chambers.

The discharge pipe of the pumps to be furnished by the engine builder and carried through the outer wall of building at least two feet beyond the wall, and to be not less than 36 inches in diameter.

The suction to be taken from a well just outside the building. The suction pipe and foot valve to be also furnished and placed by the engine builder.

The foot valve on suction pipe to be vertical and not less than 48 inches in diameter, to contain small rubber valves not exceeding 4 inches in diameter.

The aggregate area of these valves to be not less than 125 per cent of plunger area.

Steam Cylinders.—The steam cylinders to be three in number; one high, one intermediate, and one low-pressure steam cylinder, connected directly with the plungers and through a three-throw crank shaft. The steam cylinders to be fitted with Corliss valve gear, including drop cut-off valves, with dash pots. The exhaust valves on the low-pressure cylinder to be operated by separate eccentrics. All steam cylinders and cylinder heads to be steam jacketed.

The cut-off of the high-pressure steam cylinder to be controlled by a governor, and the cut-off of the other cylinders to be adjustable by hand while the engine is in motion.

The governor will be of the Porter Allen or high-speed type.

There is to be a tubular re-heater between the high and the intermediate, and between the intermediate and the low-pressure cylinders; steam of boiler pressure to circulate through the tubes of the inter-

mediate re-heater, and steam of reduced pressure through the tubes of the low-pressure re-heater.

Stuffing-boxes of Steam Cylinders.—The stuffing-boxes on steam cylinders to be not less than 8 inches deep, fitted with metallic packing of approved kind.

Indicator Gear.—A permanent gear for operating indicator for steam and pump cylinders to be attached to the engine.

Framing.—Each steam cylinder to be supported upon heavy cast-iron frames.

Crossheads.—The main crossheads to be of the best forged iron or best steel castings.

Piston Rods.—The piston rods to be of forged steel, or the best hammered iron, as may be required.

Crank Pins.—All crank pins and pins of the valve gear subject to wear, to be made of the best forged steel.

Cranks.—The cranks to be of the best hammered iron or cast-iron discs and to be forced and keyed on to the shaft; one intermediate crank to have adjustable sliding blocks.

Crank Shafts.—Crank shafts to be of the best hammered iron.

Fly Wheels.—There will be required two fly wheels of cast iron of suitable diameter and weight.

Connecting Rods.—The connecting rods to be of the best hammered iron, and to be arranged for adjustment in the bearings.

Steam Piping.—The steam piping to be furnished by the contractor and carried through outer wall of engine room; to be made of wrought-iron lap-welded tubing with heavy cast-iron flanges.

Lagging and Covering.—All parts of the steam cylinders and piping in engine room to be well covered with the best quality of non-conducting material, and neatly covered with ornamental wooden or other suitable lagging secured with brass bands, nickel plated; all attached in such a manner as to be easily removed for inspection of the various parts of the engine.

Steam Traps.—Suitable steam traps for jackets and re-heaters are to be furnished; and where possible the water of condensation is to be returned to the boilers by gravitation or otherwise.

Lubricators.—Lubricators for steam cylinders to be furnished of the positive moved type. All bearings and wearing surfaces to be furnished with drop style feed oil cups. Proper drip pans and splashers to be furnished.

Wrenches.—A full set of well-forged wrought-iron wrenches to be furnished for all bolts over one inch in diameter.

Gages and Revolution Counter.—There will be required: One steam gage for high-pressure steam cylinder; one gage for intermediate cylinder re-heater; one gage graduated to read above and below atmosphere, for the low-pressure cylinder re-heater; one gage

for the pump end; one vacuum gage and one revolution counter; all to have 8½-inch dials and to be nickel plated.

Condensing Apparatus.—There will be required an attached vertical air pump and jet condenser of ample capacity to give not less than 26 inches of vacuum, when the engine is running at a piston speed of 225 feet per minute. The condensing apparatus to be fitted to take injection water from the rising main and from pump well, as the case may require.

Air Charging Pump.—There is to be furnished an air pump ample for charging the air chambers.

Boiler-Feed Pumps.—Two boiler-feed pumps to be furnished, and to be made of double the capacity required for the engine; the speed of the boiler-feed pump, if duplex, not to exceed 50 feet per minute. One feed pump to be of the "Duplex" independent, and one attached to the engine with outside packed plungers, fitted for pumping hot water into the boilers when carrying 150 pounds of steam pressure.

Materials.—All parts of the engine and pump must be of the most suitable material for the work they are to perform, of good workmanship and neat design, and possess ample strength to render them reliable and durable. All materials to be of the best quality, all forgings to be free of faults due to welds, and all castings to be made true to form. Such parts as are liable to get out of order must be easy of access for repair or replacement. All materials to be subject to the approval of the superintendent of water works.

Painting.—Every portion of the engine and pump, except the bright and polished parts, must receive three coats of suitable paint; the last two of which shall be of a color or colors to be designated by the superintendent of water works.

Foundations.—A pile and stone foundation for these engines will be furnished by the water works division. Bidders will be required to furnish plans, with proposal submitted, showing the foundation desired by them.

Duty.—The engine will be required to perform a duty, based upon plunger displacement, equivalent to not less than 140,000,000 foot pounds of work for each 1000 pounds of commercially dry steam used,

The duty trials to be made while pumping against a pressure of 85 pounds per square inch at the pumps and with a steam pressure not exceeding 150 pounds per square inch at the throttle valve.

This duty trial to be not over twenty-four hours' duration and to be made by the superintendent of water works and the engineer of the contractor. In case of disagreement between them, the director of public works shall choose one expert, the contractor shall choose another, and these two experts shall choose the third.

These three shall determine the duty of the engine, and return their findings, which shall be final, to the director of public works. The

expenses in such a case shall be borne equally by the contracting parties.

In case the engine fails to make the required duty, the acceptance or rejection of the engine will be optional with the director of public works, and in case the engine is rejected, the city is to have the free use of the engine until another one can be procured.

The contractor will be required to guarantee the city for the period of one year after acceptance of the engine against all accidents or damages to persons or property that can be traced to or caused by defective workmanship, faulty material or design.

All proposals must be accompanied by a specification furnished by the superintendent of water works, and also by specifications and plans furnished by bidder, giving details of the construction of the engine he proposes to furnish in conformity with these specifications.

The contractor, upon the completion of the engine, shall furnish two complete sets of blue prints on cloth-backed paper of the detail drawings of the engine and appurtenances, both to be bound neatly.

No proposals will be entertained unless made on the *blanks* furnished by the superintendent of water works and delivered at the *office* of the board of control, City Hall, previous to 12 o'clock m. on the day specified.

Proposals will not be entertained for any type of engine not now in successful operation for some public water supply.

Each proposal must contain the full name of the party or parties making the same, and all persons interested therein, and must be accompanied by a bond of eight thousand ($8000) dollars of some disinterested person or persons resident of this county, or a certified check on a solvent bank in the city as surety that if the proposal be accepted a contract will be entered into.

These specifications, and the specifications, guarantees, statements and proposals that may be accepted under them, shall form a part of any contract that is entered into for the purchase of an engine; but may be changed or modified by mutual agreement between the parties.

Payments.—If requested by the contracting party, the director of public works will cause approximate estimates to be made in the following manner:

The First Estimate: To be an amount not exceeding 85 per cent of one fifth of the contract price, to be given when the steam and water cylinders are cast and machined at the works of the contractor, with pistons, plungers and rods fitted, inspected and approved by the superintendent of water works.

The Second Estimate: To be an amount not exceeding 85 per cent of one fifth of the contract price, to be given when the engine is delivered at the pumping station.

The Third Estimate: To be an amount not exceeding 85 per cent of one fifth of the contract price, to be given when the engine has been under steam for five days.

The Fourth Estimate: To be an amount not exceeding one fifth of the contract price, to be given after the duty trial and acceptance of said engine by the director of public works.

The Fifth Payment: Four months after the acceptance of the engine by the director of public works, the remaining unpaid sum, less 10 per cent of the contract price, will be paid to the contractor.

And said 10 per cent to be paid at the expiration of the one year's guarantee.

Plans and specifications for any other type of vertical triple-expansion pumping engine, in which the stroke of the steam piston is positive and unchangeable, and which will meet the requirements and give the duty stated above, may, at the option of the director of public works, be considered.

The city, by its director of public works, reserves the right to reject any or all bids, also the right to reject any type of pumping engine which in his opinion is not suitable for the existing conditions.

It is further agreed and stated as part of this specification, that in case the engine exceeds during the twenty-four hours' working test a duty of 140,000,000 foot pounds per 1000 pounds of commercially dry steam, the party of the first part, by its director of public works, will pay to the contractor for the superior efficiency of the engine, an amount to be in the ratio of $500 for each 1,000,000 foot pounds which the duty exceeds 140,000,000 foot pounds. But in case said engine fails to perform the duty of 140,000,000 foot pounds per 1000 pounds of commercially dry steam, as hereinbefore specified, during the working test of twenty-four hours, the said party of the first part, by its director of public works, shall have the right, at his option, and if he deems best so to do, to retain said engine and deduct from the contract price an amount to be in ratio of $1500 for each 1,000,000 foot pounds which the duty falls below said 140,000,000 foot pounds, as the agreed difference in value.

The following are the specifications for a modern, high-speed, compound engine of the type shown in Fig. 194.

Specifications for Engine from Engine Company.

Style of Engine.—One side-crank tandem compound automatic enclosed engine for direct connection to 125 kw. direct-current generator, including extended shaft and outboard bearing and sub-base under engine and generator.

Size.—Diameter of high-pressure cylinder, 11 inches; diameter of

low-pressure cylinder, 18 inches; stroke, 16 inches; revolutions per minute, 250.

Power.—Steam pressure maintained at the throttle valve of engine, not less than 150 pounds, non-condensing. Economical rating, 184 indicated horse power.

Dimensions.—Governor wheel, diameter 66 inches; width of face 14 inches; diameter of steam pipe 4 inches, exhaust pipe 9 inches; floor space, length 16 feet 3 inches, width 10 feet 3 inches; shipping weight, approximately 25,650 pounds; main bearing, diameter 7¾ inches, length 14 inches; crank pin, diameter 5¼ inches, length 3⅝ inches; crosshead pin, diameter 3⅝ inches, length 3⅝ inches; area crosshead shoe, 90 square inches; piston rod, diameter 2⅜ inches.

Workmanship and Materials.—Iron castings will be of such grade as to insure greatest strength, best wearing qualities, and superior finish.

We guarantee the workmanship and materials to be of the best quality, and will replace free of charge, on cars, any part that may fail on account of defective workmanship and materials, within one year after shipment of the engine from our works, but no claim for labor or damages will be allowed.

All parts will be made to gage, and interchangeable as far as possible, and all working parts having flat surfaces will be scraped to surface plates.

All moving parts will be carefully balanced, and the engine will operate quietly and without undue vibration.

Cylinders.—Cylinders will be made of special hard, close-grained cast iron, accurately bored and finished to micrometer measurements; and will be neatly covered with metal lagging inclosing a layer of non-conducting covering.

Oiling System.—Crank pin, journal, crosshead, crosshead pin, eccentric and governor suspension pin will be automatically and copiously lubricated, reducing friction to a minimum.

Crank Shaft and Main Bearings.—Crank shaft will be a single forging of open hearth steel containing the proper amount of carbon to give exceptional strength, and will be ground to smooth and accurate running surfaces.

Main bearings are adjustable, and of the self-oiling type, having removable shells lined with best hard babbitt.

Regulation.—The regulation will be by means of a governor of the inertia type. The speed of the engine will not vary over 1½ per cent at all loads and pressures within the rated capacity of the engine.

Wheels.—The wheels will be strong, well proportioned, with a large margin for safety; having flanges on both edges of the rim, and will be carefully balanced and finished.

Painting.—Manner of painting will be as follows: One coat prim-

ing, followed by one coat filler well rubbed down; then one coat standard steel color with varnish, finally finishing coat steel color.

Foundation Plans.—Foundation plans to cover the special conditions required will be furnished.

Fittings.—The engine will be provided with screwed throttle valve, exhaust flange, lubricator, drop-forged case-hardened wrenches, special wrenches and spanners, indicator plugs, drip cocks, foundation bolts, plates and nuts.

DIRECT CONNECTION TO DYNAMO.

Shaft.—The crank shaft will be extended to carry the armature of dynamo, and to form an outer journal.

Standard Sizes.—The portion upon which the armature is mounted will be finished to conform to the drawings and gages to be furnished by the dynamo builders,—provided that the diameter of shaft in armature bore shall not exceed that recommended for engines of this size by the joint committees on standardization appointed by the Mechanical and Electrical Engineering Societies,—except at an extra price. These sizes are as follows:

Dynamo Kilowatts	25	35	50	75	100	150	200
Diameter shaft center crank engines, inches	4	4	4½	5½	6	7	8
Diameter shaft side crank engines, inches	4½	5½	6½	7½	8½	10	11

Key.—Armature key will be furnished in accordance with dynamo drawings.

Sub-Base Extension.—Sub-base will be extended to carry field frame of dynamo, and the pedestal supporting outer bearing.

The top of this extension will be of suitable form to carry generator. It will be planed, and holes will be drilled and tapped in it, in accordance with drawings to be furnished by the dynamo builders.

Outboard Bearing.—We will furnish our regular style pedestal, fitted with ring oiling bearing; pedestal to be mounted on outer end of sub-base extension and bolted to it.

Parts Pertaining to Dynamo.—Any parts which the special design of generator makes necessary, it is understood are to be furnished by the dynamo builders; or if furnished by us, an extra charge to cover cost of same will be made.

If requested, the crank shaft will be shipped to the generator builder to have armature pressed on,—with the understanding that the purchaser is to pay the freight on the shaft and assume all risk of damage in transit or otherwise.

Delivery.—Delivery to be made f.o.b. cars, within .... from acceptance of this proposition with all data, such as steam pressure, speed of engine, size of wheels, approved dynamo drawings if direct connected, or any other necessary information to enable us to adapt the engine to the place or character of service required.

The time of delivery specified is given in good faith, but is contingent upon strikes, accidents, fire, flood, inability to procure labor or materials, and all other causes beyond our control.

If by request of the purchaser shipment of the engine be delayed more than three days after the date of delivery specified, it is agreed that the engine shall be placed in storage at his risk and expense. It is further agreed in case of delay on purchaser's account as aforesaid, that all payments on account of contract price shall date from the day the engine is ready to ship, and not from the date of actual delivery.

Freights.—When delivered prices are named on engines, freight is to be paid by the purchaser, who will be given credit for the same on account.

Upon each shipment this company agrees to mail purchaser bills of lading, showing the number of parts, boxes, crates, pieces of machinery, etc., that have been shipped, and the purchaser will compare shipment as delivered with these bills of lading, and in case shortage is found, he will receive from carrier only under a protest—claim for such pieces or parts as are missing or damaged.

The delivery of said goods at place designated, without objection, or their receipt by consignee, shall constitute a waiver of any damages claimed on account of any delay.

Expert.—If desired by purchaser, we will furnish the time of our expert to superintend erection and starting of engines, for which purchaser agrees to pay five dollars per day from the time he leaves our works until his return, also railroad fare, board and expenses, or in case this contract includes the erecting and starting of engines by this company, the mechanic sent for such purpose shall report to the purchaser or his engineer as soon as the engines are ready for operation, and after the same shall have been operated or ready to operate under the conditions of the specifications for a period of three days, the purchaser hereby agrees to withdrawal of our mechanic from further duty, and the purchaser will assume all future control of operation and hereby agrees to the acceptance of the engines so far as their operation is concerned without further demands excepting such as are provided for him in the usual structural guarantee.

Delivery on Cars Only.—If this proposition is accepted by the purchaser for the engine delivered free on cars only, an expert to start not being included, it is agreed that the purchaser assumes all risks and waives all claims upon this company for unsatisfactory operation.

Foundations.—If we are to deliver engine on foundation, the purchaser undertakes to furnish sufficient openings to get engine to foundation, and to inform us as to points of location, floor lines, etc., before our work is begun; and further, to pay extra for labor and material in making changes thereafter ordered. If we are to furnish foundation, it is understood that said foundation is to be built to our standard foundation plans, and should the conditions of the ground where located, or the floor level, be such that an extra depth of foundation is required, the purchaser agrees to pay for said extra depth at cost, plus 10 per cent.

Overloading.—The purchaser undertakes that the engine shall not be taxed to run at a greater pressure, speed or power than it is rated to operate, unless at his own risk, and that our directions as to its operation and management shall be carefully observed.

Safe Keeping.—The purchaser shall become responsible for the safe keeping, and shall insure this machinery for our benefit, as our interest may appear, from the time the same shall be delivered upon the designated premises; any loss or damage to parts to be borne by purchaser.

Title to Plant.—The title to this machinery shall not vest in the purchaser until it is fully paid for, and the fact that said machinery may be attached to realty, by any means whatever, shall not be considered as making it a part of such realty, but till fully paid for, the same shall be and remain personal property, and the purchaser undertakes to execute or cause to be executed, acknowledge and deliver to us all legal instruments necessary and appropriate to preserve our title therein. If notes are accepted, the title to said machinery shall not pass from us until such notes, and all extensions and renewals thereto, shall have been paid. And if default is made in the payment of contract price of said machinery, whether evidenced by note or otherwise, the right is expressly reserved and given to enter the premises where said machinery may be and remove the same as our property.

Taking Other Security.—The taking of security other than provided for shall not operate as a waiver of our statutory lien, and consent is given that we may, without notice, accept security, and thereafter increase, diminish, exchange, or release the same.

Payment.—The contract price, or any unpaid part thereof, shall draw interest after due at the legal rate until paid; and when the time

of payment extends beyond thirty days, such deferred payments shall be evidenced by negotiable note, at our option.

Agreements not Expressed.—There are no understandings or agreements not expressed herein, and nothing is included that is not particularly mentioned.

169. Cost of Engines.—In 1902 the author obtained figures from different builders showing the selling prices, f.o.b. the factory, of various classes of engines, together with the approximate cost of the foundations.

These prices were plotted as ordinates, using the rated horse powers as abscissas, and curves were drawn. The equations of straight lines which represent the curves approximately are as follows:—

Type.	*Cost.*
Simple high-speed engines,	300 + 8 h.p. dollars.
Setting high-speed engines,	50 + 0.75 h.p. dollars.
Compound high-speed engines,	1000 + 15 h.p. dollars.
Simple low-speed engines,	1000 + 10 h.p. dollars.
Compound low-speed engines,	2000 + 13 h.p. dollars.
Setting low-speed engines,	500 + 1.3 h.p. dollars.

In 1906 a more comprehensive investigation was made under the direction of the author by Messrs. Baker and Ingersoll, senior students in the Case School; the figures given, as in the former case, representing the price of the engines delivered on the cars at the factory.

The prices in the subjoined tables represent in each case an average of several quotations from different builders.

TABLE VI.

PRICES OF SIMPLE NON-CONDENSING ENGINES.

Low Speed.		High Speed.	
I.H.P.	Price.	I.H.P.	Price.
50	$ 780	50	$ 825
75	1050	60	790
		75	933
80	1350	80	920
90	1400	90	1215
100	1307	100	1100
115	1600	120	1160
125	2035	125	1260
150	1500	150	1412
160	2637	160	1400
175	2085	175	1740
185	2100	200	1734
200	2725	225	1840
250	2977	250	2200
290	3837	275	2410
315	4200	300	2400
		325	2500
Average of ten quotations.		Average of twelve quotations.	

TABLE VII.

PRICES OF HORIZONTAL CROSS-COMPOUND ENGINES (CONDENSING.)

Low Speed.		High Speed.	
I.H.P.	Price.	I.H.P.	Price.
225	$ 6500	80	$1400
250	6100	85	1900
285	7500	115	2560
300	6650	138	2790
375	8200	160	2070
400	8100	175	3380
410	9000	204	3027
482	10000	250	3157
500	9100	325	3550
600	10300	395	5100
616	11500	400	3700
697	12500	460	6800
750	12000	560	7650
875	15000		
1000	17500		
Average of ten quotations.		Average of seven quotations.	

TABLE VIII.

PRICES OF TANDEM, COMPOUND, HIGH SPEED ENGINES.

	Price.
72	$2175
96	2360
122	2575
145	2830
160	2900
186	3165
200	3250
215	3690
270	4360
310	4410
325	4650
400	5050
500	7350
600	8100
750	9800

Average of five quotations.

TABLE IX.

PRICES OF VERTICAL ENGINES.

Simple High Speed.		Cross-compound High Speed.	
I.H.P.	Price.	I.H.P.	Price.
10	$ 255	50	$1050
20	285	75	1500
30	375	100	1457
40	562	150	2237
50	720	200	2350
60	720	250	3825
75	830	300	4380
90	900	350	5200
100	1100	400	5600
120	1900	450	5630
300	3050	500	5550
Average of four quotations.		Average of four quotations.	

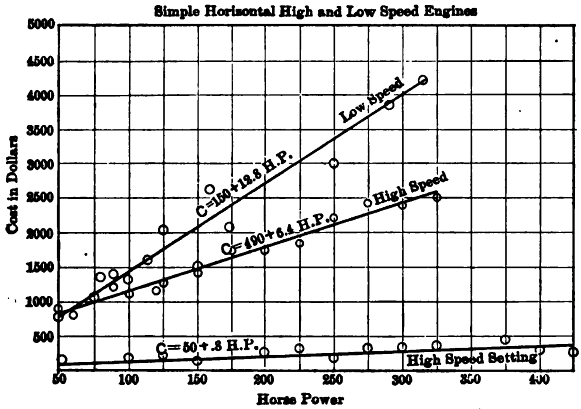


Fig. 195. Diagram of Costs, Simple Non-Condensing Engines.

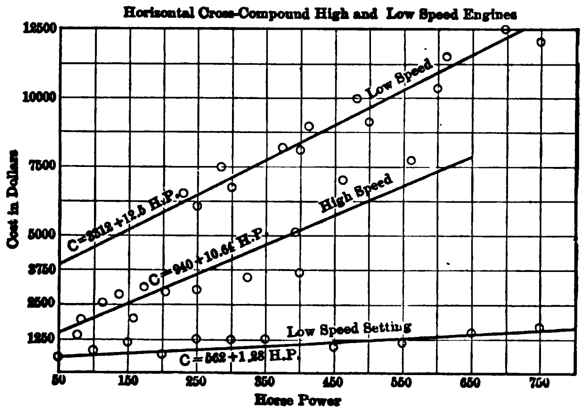


Fig. 196. Diagram of Costs, Compound Engines. (Condensing.)

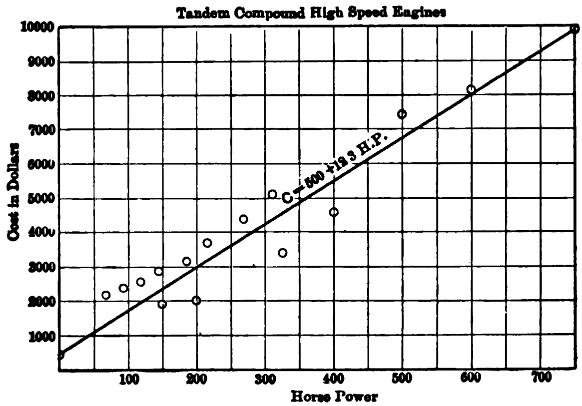


Fig. 197. Diagram of Costs, Tandem High-Speed Engines.

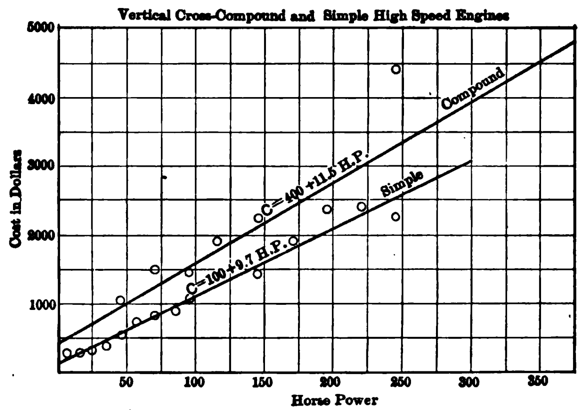


Fig. 198. Diagram of Costs, Vertical High-Speed Engines.

An examination of the tables shows considerable irregularity in consecutive prices. For instance, in Table VI. the price of 100 h.p. engine is less than that of 90 h.p. engine, the price of 150 h.p. less than that of 125 h.p., and so on.

This is sometimes due to the difference in cost of standard and of special sizes, but in many cases it is accidental and shows

that only average values can be obtained and that there is liable to be a wide variation from these in particular cases. Figs. 195 to 198 are plotted from the foregoing tables. The equations of the straight lines are as follows for the various classes of engines represented:

SELLING PRICES OF ENGINES IN TERMS OF THE RATED HORSE POWER.

	Price in Dollars.
Simple Horizontal Low-speed Engines	150 + 12.84 H.P.
Simple Horizontal High-speed Engines	490 + 6.4 H.P.
Cross-compound Horizontal Low-speed Engines	3312 + 12.5 H.P.
Cross-compound Horizontal High-speed Engines	940 + 10.64 H.P.
Tandem Compound Horizontal High-speed Engines	500 + 12.3 H.P.
Cross-compound Vertical High-speed Engines	400 + 11.5 H.P.
Simple Vertical High-speed Engines	100 + 9.7 H.P.

170. Choice of an Engine.—In determining upon the most economical class of engine for a given plant, several considerations must be taken into account. The question of condensation vs. non-condensation is usually determined by the location and the availability of water. It is only in large installations running night and day that the use of cooling towers can be considered.

The choice of high or low speed depends largely on the type of transmission employed. Small units directly connected to electric generators are usually high speed on account of the smaller size and lower cost of the high-speed generators.

Here again, it is only in large plants running continuously that low-speed multi-polar generators can be economically used, where the daily saving in steam will more than pay the interest on the larger investment.

The use of tandem compounds and of vertical engines is largely a question of floor space. The tandem compound is also in a way an intermediate step between the simple engine and the elaborate cross-compound, more economical than the former and not so expensive as the latter.

It is also available in smaller sizes than the cross-compound.

APPENDIX

TABLE X.

Properties of Saturated Steam.

Absolute pressure lb. per sq. in.	Temperature of boiling point deg. Fahr.	Heat of the liquid from 32 deg. Fahr.	Total heat from 32 deg. Fahr.	Latent heat	Weight of a cu. ft. in pounds	Cubic feet per pound	Entropy of liquid	Entropy of steam
p	t	q	H	L	d	$1 \div d$	θ	ϕ
1.0	102.0	70.0	1113.1	1043.0	0.00299	334.6	0.133	1.853
2.0	126.3	94.4	1120.5	1026.1	0.00576	173.6	0.175	1.748
3.0	141.6	109.8	1125.1	1015.3	0.00844	118.4	0.201	1.684
4.0	153.1	121.4	1128.6	1007.2	0.01107	90.31	0.220	1.640
5.0	162.3	130.7	1131.5	1000.8	0.01366	73.22	0.235	1.606
6.0	170.1	138.6	1133.8	995.2	0.01622	61.67	0.248	1.577
7.0	176.9	145.4	1135.9	990.5	0.01874	53.37	0.258	1.553
8.0	182.9	151.5	1137.7	986.2	0.02112	47.07	0.268	1.531
9.0	188.3	156.9	1139.4	982.5	0.02374	42.13	0.277	1.513
10.0	193.2	161.9	1140.9	979.0	0.02621	38.16	0.284	1.496
11.0	197.8	166.5	1142.3	975.8	0.02866	34.88	0.291	1.480
12.0	202.0	170.7	1143.6	972.9	0.03111	32.14	0.297	1.467
13.0	205.9	174.6	1144.7	970.1	0.03355	29.82	0.303	1.455
14.0	209.6	178.3	1145.8	967.5	0.03600	27.79	0.309	1.443
14.7	212.0	180.7	1146.6	965.8	0.03758	26.64	0.312	1.435
15.0	213.0	181.8	1146.9	965.1	0.03826	26.15	0.314	1.432
16.0	216.3	185.1	1147.9	962.8	0.04067	24.59	0.319	1.421
17.0	219.4	188.3	1148.9	960.6	0.04307	23.22	0.324	1.412
18.0	222.4	191.3	1149.8	958.5	0.04547	22.00	0.328	1.402
19.0	225.2	194.1	1150.7	956.6	0.04786	20.90	0.332	1.394
20.0	227.9	196.9	1151.5	954.6	0.05023	19.91	0.336	1.385
21.0	230.5	199.5	1152.3	952.8	0.05259	19.01	0.340	1.378
22.0	233.1	202.0	1153.0	951.0	0.05495	18.20	0.344	1.370
23.0	235.5	204.5	1153.7	949.2	0.05731	17.45	0.347	1.363
24.0	237.8	206.8	1154.4	947.6	0.05966	16.76	0.350	1.356
25.0	240.0	209.1	1155.1	946.0	0.06199	16.13	0.354	1.350
26.0	242.2	211.2	1155.8	944.6	0.06432	15.55	0.357	1.343
27.0	244.3	213.4	1156.5	943.1	0.06666	15.00	0.360	1.337
28.0	246.4	215.4	1157.1	941.7	0.06899	14.49	0.363	1.331
29.0	248.3	217.4	1157.7	940.3	0.07130	14.03	0.365	1.325
30.0	250.3	219.4	1158.3	938.9	0.07360	13.59	0.368	1.320
31.0	252.1	221.3	1158.8	937.5	0.07590	13.18	0.371	1.315
32.0	254.0	223.1	1159.4	936.3	0.07821	12.78	0.373	1.310
33.0	255.8	224.9	1159.9	935.0	0.08051	12.41	0.376	1.305
34.0	257.5	226.7	1160.4	933.7	0.08280	12.07	0.378	1.300
35.0	259.2	228.4	1161.0	932.6	0.08508	11.75	0.381	1.295

PROPERTIES OF SATURATED STEAM (Continued).

Absolute pressure lb. per sq. in.	Temperature of boiling point deg. Fahr.	Heat of the liquid from 32 deg. Fahr.	Total heat from 32 deg. Fahr.	Latent heat	Weight of a cu. ft. in pounds	Cubic feet per pound	Entropy of liquid	Entropy of steam
p	t	q	H	L	d	$1 \div d$	θ	ϕ
40.0	267.1	236.4	1163.4	927.0	0.09644	10.37	0.392	1.273
45.0	274.3	243.6	1165.6	922.0	0.1077	9.287	0.402	1.254
50.0	280.8	250.2	1167.6	917.4	0.1188	8.414	0.411	1.236
55.0	286.9	256.3	1169.4	913.1	0.1299	7.696	0.419	1.221
60.0	292.5	261.9	1171.2	909.3	0.1409	7.096	0.426	1.206
65.0	297.8	267.2	1172.7	905.5	0.1519	6.583	0.433	1.193
70.0	302.7	272.2	1174.3	902.1	0.1628	6.144	0.440	1.181
75.0	307.4	276.9	1175.7	898.8	0.1736	5.762	0.446	1.169
80.0	311.8	281.4	1177.0	895.6	0.1843	5.425	0.452	1.159
85.0	316.0	285.8	1178.3	892.5	0.1951	5.125	0.458	1.149
90.0	320.0	290.0	1179.6	889.6	0.2058	4.858	0.463	1.139
95.0	323.9	294.0	1180.7	886.7	0.2165	4.619	0.468	1.129
100.0	327.6	297.9	1181.9	884.0	0.2271	4.403	0.473	1.121
105.0	331.1	301.6	1182.9	881.3	0.2378	4.206	0.478	1.112
110.0	334.6	305.2	1184.0	878.8	0.2484	4.026	0.482	1.105
115.0	337.9	308.7	1185.0	876.3	0.2589	3.862	0.486	1.097
120.0	341.0	312.0	1186.0	874.0	0.2695	3.711	0.491	1.089
125.0	344.1	315.2	1186.9	871.7	0.2800	3.572	0.495	1.082
130.0	347.1	318.4	1187.8	869.4	0.2904	3.444	0.498	1.076
135.0	350.0	321.4	1188.7	867.3	0.3009	3.323	0.502	1.070
140.0	352.8	324.4	1189.5	865.1	0.3113	3.212	0.506	1.063
145.0	355.6	327.2	1190.4	863.2	0.3218	3.107	0.509	1.057
150.0	358.3	330.0	1191.2	861.2	0.3321	3.011	0.513	1.051
155.0	360.9	332.7	1192.0	859.3	0.3426	2.919	0.516	1.045
160.0	363.4	335.4	1192.8	857.4	0.3530	2.833	0.519	1.040
165.0	365.9	338.0	1193.6	855.6	0.3635	2.751	0.522	1.035
170.0	368.3	340.5	1194.3	853.8	0.3737	2.676	0.525	1.030
175.0	370.6	343.0	1195.0	852.0	0.3841	2.603	0.528	1.025
180.0	373.0	345.4	1195.7	850.3	0.3945	2.535	0.531	1.020
185.0	375.23	347.8	1196.4	848.6	0.4049	2.470	0.534	1.015
190.0	377.4	350.1	1197.1	847.0	0.4153	2.408	0.537	1.010
195.0	379.6	352.4	1197.7	845.3	0.4257	2.349	0.539	1.006
200.0	381.7	354.6	1198.4	843.8	0.4359	2.294	0.542	1.002
250.0	401.0	374.7	1204.2	829.5	0.5393	1.854	0.566	0.963
300.0	417.4	391.9	1209.3	817.4	0.6440	1.554	0.586	0.930
400.0	444.9	419.8	1217.7	797.9	0.8572	1.167	0.618	0.880

TABLE XI.

WEIGHT OF WATER.

Temperature deg. Fahr.	Heat units per pound	Weight pounds per cu. ft.	Temperature deg. Fahr.	Heat units per pound	Weight pounds per cu. ft.	Temperature deg. Fahr.	Heat units per pound	Weight pounds per cu. ft.
32	0.00	62.42	123	91.09	61.68	169	137.46	60.79
35	3.02	62.42	124	92.10	61.67	170	138.46	60.77
40	8.06	62.42	125	93.10	61.65	171	139.47	60.75
45	13.08	62.42	126	94.11	61.63	172	140.48	60.73
50	18.10	62.41	127	95.12	61.61	173	141.49	60.70
52	20.11	62.40	128	96.13	61.60	174	142.50	60.68
54	22.11	62.40	129	97.14	61.58	175	143.50	60.66
56	24.11	62.39	130	98.14	61.56	176	144.51	60.64
58	26.12	62.38	131	99.15	61.54	177	145.52	60.62
60	28.12	62.37	132	100.16	61.52	178	146.53	60.59
62	30.12	62.36	133	101.17	61.51	179	147.54	60.57
64	32.12	62.35	134	102.18	61.49	180	148.54	60.55
66	34.12	62.34	135	103.18	61.47	181	149.55	60.53
68	36.12	62.33	136	104.19	61.45	182	150.56	60.50
70	38.11	62.31	137	105.20	61.43	183	151.57	60.48
72	40.11	62.30	138	106.21	61.41	184	152.58	60.46
74	42.11	62.28	139	107.22	61.39	185	153.58	60.44
76	44.11	62.27	140	108.22	61.37	186	154.59	60.41
78	46.10	62.25	141	109.23	61.36	187	155.60	60.39
80	48.09	62.23	142	110.24	61.34	188	156.61	60.37
82	50.08	62.21	143	111.25	61.32	189	157.62	60.34
84	52.07	62.19	144	112.26	61.30	190	158.62	60.32
86	54.06	62.17	145	113.26	61.28	191	159.63	60.29
88	56.05	62.15	146	114.27	61.26	192	160.63	60.27
90	58.04	62.13	147	115.28	61.24	193	161.64	60.25
92	60.03	62.11	148	116.29	61.22	194	162.65	60.22
94	62.02	62.10	149	117.30	61.20	195	163.66	60.20
96	64.01	62.07	150	118.30	61.18	196	164.66	60.17
98	66.01	62.05	151	119.31	61.16	197	165.67	60.15
100	68.01	62.02	152	120.32	61.14	198	166.68	60.12
102	70.00	62.00	153	121.33	61.12	199	167.69	60.10
104	72.00	61.97	154	122.34	61.10	200	168.70	60.07
106	74.00	61.95	155	123.34	91.08	201	169.70	60.05
108	76.00	61.92	156	124.35	61.06	202	170.71	60.02
110	78.00	61.89	157	125.36	61.04	203	171.72	60.00
112	80.00	61.86	158	126.37	61.02	204	172.73	59.97
113	81.01	61.84	159	127.38	61.00	205	173.74	59.95
114	82.02	61.83	160	128.38	60.98	206	174.74	59.92
115	83.02	61.82	161	129.39	60.96	207	175.75	59.89
116	84.03	61.80	162	130.40	60.94	208	176.76	59.87
117	85.04	61.78	163	131.41	60.92	209	177.77	59.84
118	86.05	61.77	164	132.42	60.90	210	178.78	59.82
119	87.06	61.75	165	133.42	60.87	211	179.78	59.79
120	88.06	61.74	166	134.43	60.85	212	180.79	59.76
121	89.07	61.72	167	135.44	60.83			
122	90.08	61.70	168	136.45	60.81			

TABLE XII.

THE PROPERTIES OF ANHYDROUS AMMONIA.

Temperature deg. Fahr (t)	Pressure (Absolute) lb. per sq. in. (p)	Heat of Vaporization Thermal Units (L)	External Heat in B.t.u $\left(\frac{pu}{778}\right)$	Volume of Vapor per lb., cu. ft. (v)	Volume of Liquid per lb. cu. ft. (v_s)
− 40	10.69	579.67	48.25	24.38	.0234
− 35	12.31	576.69	48.35	21.21	.0236
− 30	14.13	573.69	48.85	18.67	.0237
− 25	16.17	570.68	49.16	16.42	.0238
− 20	18.45	567.67	49.44	14.48	.0240
− 15	20.99	564.64	49.74	12.81	.0242
− 10	23.77	561.61	50.05	11.36	.0243
− 5	27.57	558.56	50.44	9.89	.0244
∓ 0	30.37	555.50	51.38	9.14	.0246
+ 5	34.17	552.43	50.84	8.04	.0247
+ 10	38.55	549.35	51.13	7.20	.0249
+ 15	42.93	546.26	51.33	6.46	.0250
+ 20	47.95	543.15	51.65	5.82	.0252
+ 25	53.43	540.03	51.81	5.24	.0253
+ 30	59.41	536.92	52.02	4.73	.0254
+ 35	65.93	533.78	52.22	4.28	.0256
+ 40	73.00	530.63	52.42	3.88	.0257
+ 45	80.66	527.47	52.62	3.53	.0260
+ 50	88.96	524.30	52.82	3.21	.02601
+ 55	97.93	521.12	53.01	2.93	.02603
+ 60	107.60	517.93	53.21	2.67	.0265
+ 65	118.03	515.33	53.40	2.45	.0266
+ 70	129.21	511.52	53.67	2.24	.0268
+ 75	141.25	508.29	53.76	2.05	.0270
+ 80	154.11	504.66	53.96	1.89	.0272
+ 85	167.86	501.81	54.15	1.74	.0273
+ 90	182.80	498.11	54.28	1.61	.0274
+ 95	198.37	495.29	54.41	1.48	.0276
+ 100	215.14	491.50	54.54	1.36	.0277

TABLE XIII.

Table of Hyperbolic Logarithms ($\log_e$).

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.05	0.049	2.65	.975	4.25	1.447	5.85	1.766	7.45	2.008
1.10	.095	2.70	.993	4.30	1.459	5.90	1.775	7.50	2.015
1.15	.140	2.75	1.012	4.35	1.470	5.95	1.783	7.55	2.022
1.20	.182	2.80	1.030	4.40	1.482	6.00	1.792	7.60	2.028
1.25	.223	2.85	1.047	4.45	1.493	6.05	1.800	7.65	2.035
1.30	.262	2.90	1.065	4.50	1.504	6.10	1.808	7.70	2.041
1.35	.300	2.95	1.082	4.55	1.515	6.15	1.816	7.75	2.048
1.40	.336	3.00	1.099	4.60	1.526	6.20	1.824	7.80	2.054
1.45	.372	3.05	1.115	4.65	1.537	6.25	1.833	7.85	2.061
1.50	.405	3.10	1.131	4.70	1.548	6.30	1.841	7.90	2.067
1.55	.438	3.15	1.147	4.75	1.558	6.35	1.848	7.95	2.073
1.60	.470	3.20	1.163	4.80	1.569	6.40	1.856	8.00	2.079
1.65	.500	3.25	1.179	4.85	1.579	6.45	1.864	8.05	2.086
1.70	.531	3.30	1.194	4.90	1.589	6.50	1.872	8.10	2.092
1.75	.560	3.35	1.209	4.95	1.599	6.55	1.879	8.15	2.098
1.80	.588	3.40	1.224	5.00	1.609	6.60	1.887	8.20	2.104
1.85	.612	3.45	1.238	5.05	1.619	6.65	1.895	8.25	2.110
1.90	.642	3.50	1.253	5.10	1.629	6.70	1.902	8.30	2.116
1.95	.668	3.55	1.267	5.15	1.639	6.75	1.910	8.35	2.122
2.00	.693	3.60	1.281	5.20	1.649	6.80	1.917	8.40	2.128
2.05	.718	3.65	1.295	5.25	1.658	6.85	1.924	8.45	2.134
2.10	.742	3.70	1.308	5.30	1.668	6.90	1.931	8.50	2.140
2.15	.765	3.75	1.322	5.35	1.677	6.95	1.939	8.55	2.146
2.20	.788	3.80	1.335	5.40	1.686	7.00	1.946	8.60	2.152
2.25	.811	3.85	1.348	5.45	1.696	7.05	1.953	8.65	2.158
2.30	.833	3.90	1.361	5.50	1.705	7.10	1.960	8.70	2.163
2.35	.854	3.95	1.374	5.55	1.714	7.15	1.967	8.75	2.169
2.40	.875	4.00	1.386	5.60	1.723	7.20	1.974	8.80	2.175
2.45	.896	4.05	1.399	5.65	1.732	7.25	1.981	8.85	2.180
2.50	.916	4.10	1.411	5.70	1.740	7.30	1.988	8.90	2.186
2.55	.936	4.15	1.423	5.75	1.749	7.35	1.995	8.95	2.192
2.60	.956	4.20	1.435	5.80	1.758	7.40	2.001	9.00	2.198

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